COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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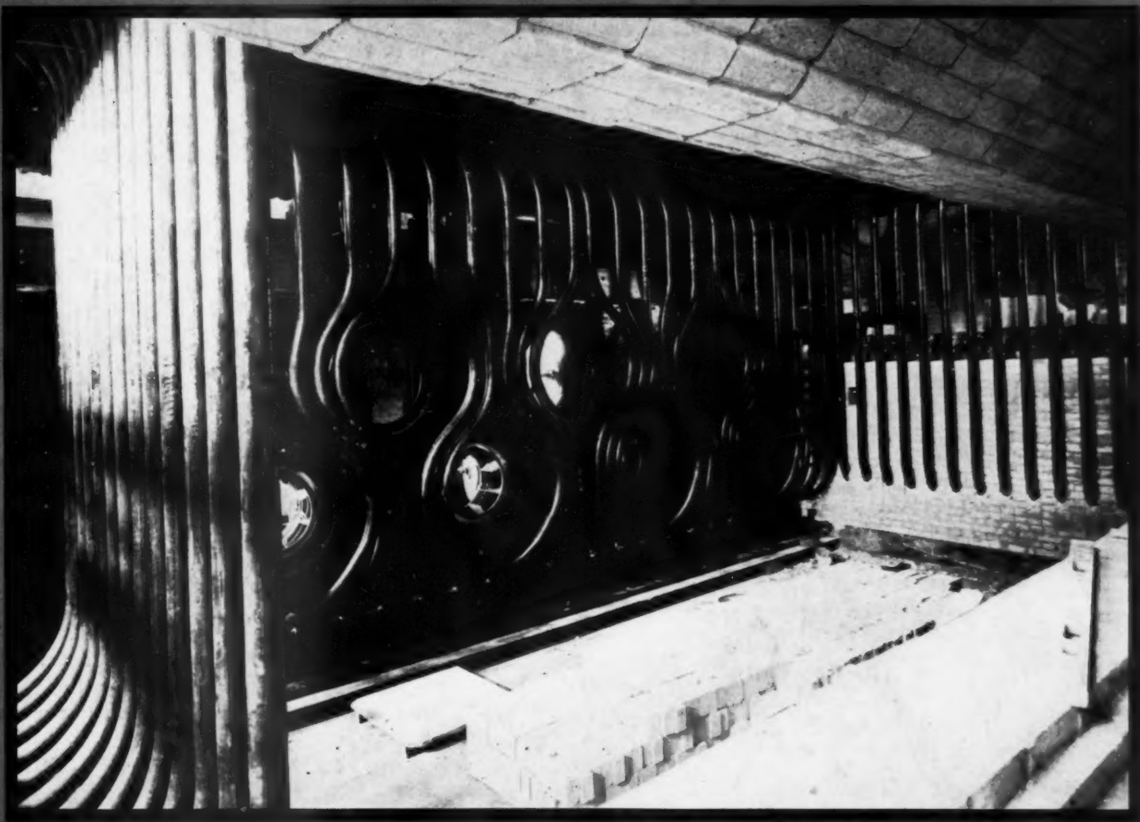


Note how the front wall tubes are bent around the burners in this boiler now nearing completion at the plant of the Pacific Lumber Co., Scotia, Calif.

Coal Sampling Methods of The Detroit Edison Company

Strength and Flexibility of Corrugated and Creased-Bend Piping

83.48% Efficiency★

with C-E Multiple Retort Stokers at Washington Central Heating Plant

IN JANUARY, 1934, the new central heating plant for government buildings in Washington, D. C., was completed and placed in service, the boiler equipment consisting of six complete C-E units as described below.

The U. S. Treasury Department specifications for these units called for an efficiency of 82 per cent at 129,000 lb per hr equivalent evaporation and included a bonus and penalty clause.

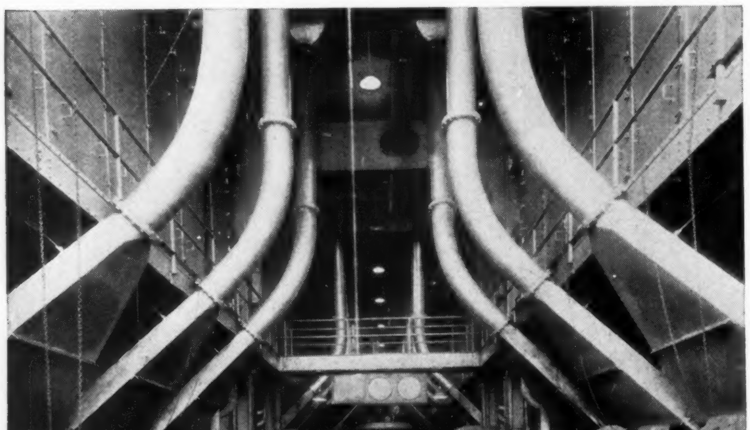
Tests to determine the ability of the equipment to meet the performance specified were begun in the early part of this year. The principal data of seven official tests are shown in the accompanying table. The first column shows the averaged results of five tests at 132,497 lb per hr equivalent evaporation and the second the averaged results of two tests at 222,209 lb per hr equivalent evaporation. These tests were made with weighed water and coal in accordance with the A.S.M.E. Power Test Code for stationary steam generating units. They were conducted under the supervision of the U. S. Bureau of Mines.

As Item 13 of the test data indicates, the results exceeded the performance specified and accordingly a bonus was awarded.

In addition to these tests, a 5-day endurance trial was made during which a steam output of 220,000 lb per hr was maintained with 2-hr peaks of 240,000 lb per hr every 24 hours. Thorough inspection of the stoker at the end of this run showed it to be in perfect condition.

The ability of these C-E Multiple Retort Stokers to give low cost as well as dependable and highly efficient service is evidenced by plant records for the past winter heating season (Oct. 1, 1934 to Mar. 31, 1935), which show the maintenance per ton of coal burned for all six stokers to be virtually nil.

These results speak for themselves. They point clearly to one conclusion—*that this latest design of C-E Multiple Retort Stoker is today's outstanding value in the multiple retort stoker field.* . . . Combustion Engineering Company, Inc., 200 Madison Avenue, New York.



Boiler Room, Central Heating Plant, Washington, D. C. Installation includes six complete C-E units comprised of C-E Sectional Header Boilers, C-E Multiple Retort Stokers and C-E Furnaces, partially water cooled. Present operating pressure—200 lb (maximum operating pressure—400 lb). At 200 lb pressure, rated maximum capacity—237,000 lb equivalent; rated continuous capacity—215,000 lb equivalent.

★Boiler and Furnace Efficiency only —No Air Preheater, No Economizer

PRINCIPAL DATA OF TESTS

		Averaged Results (5 Tests)	Averaged Results (2 Tests)
1.	Duration, hr.	22.01	12.09
2.	Water evaporation per hr—actual, lb.	**116,022**	**194,827**
3.	Water equivalent per hr., lb.	**132,497**	**222,209**
4.	Coal fired per hr., lb.	11,436	19,862
	Coal Analyses (as fired)		
5.	Carbon, %	76.66	76.12
6.	Hydrogen, %	4.50	4.54
7.	Moisture, %	3.15	3.66
8.	Ash, %	10.06	10.31
9.	Volatile Matter, %	24.60	24.40
10.	Fixed Carbon, %	62.19	61.63
11.	Heat Value, Btu	13,520	13,370

Heat Balance—Per Cent

12.	Heat value of coal	100.00	100.00
13.	Heat absorbed by boiler	**83.48**	**81.42**
14.	Heat lost	16.52	18.58
15.	Moisture in coal	0.29	0.35
16.	Water from burning H_2	3.67	3.89
17.	Dry stack gases	9.72	11.52
18.	CO in stack gases	0.08	0.07
19.	Combustible in flue dust	0.03	0.05
20.	Combustible in cinders	0.09	0.53
21.	Combustible in ash	1.85	1.37
22.	Radiation and unaccounted.	0.79	0.80

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COMBUSTION ENGINEERING

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME SEVEN NUMBER FOUR

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FOR OCTOBER 1935

FEATURE ARTICLES

EDITORIALS

DEPARTMENTS

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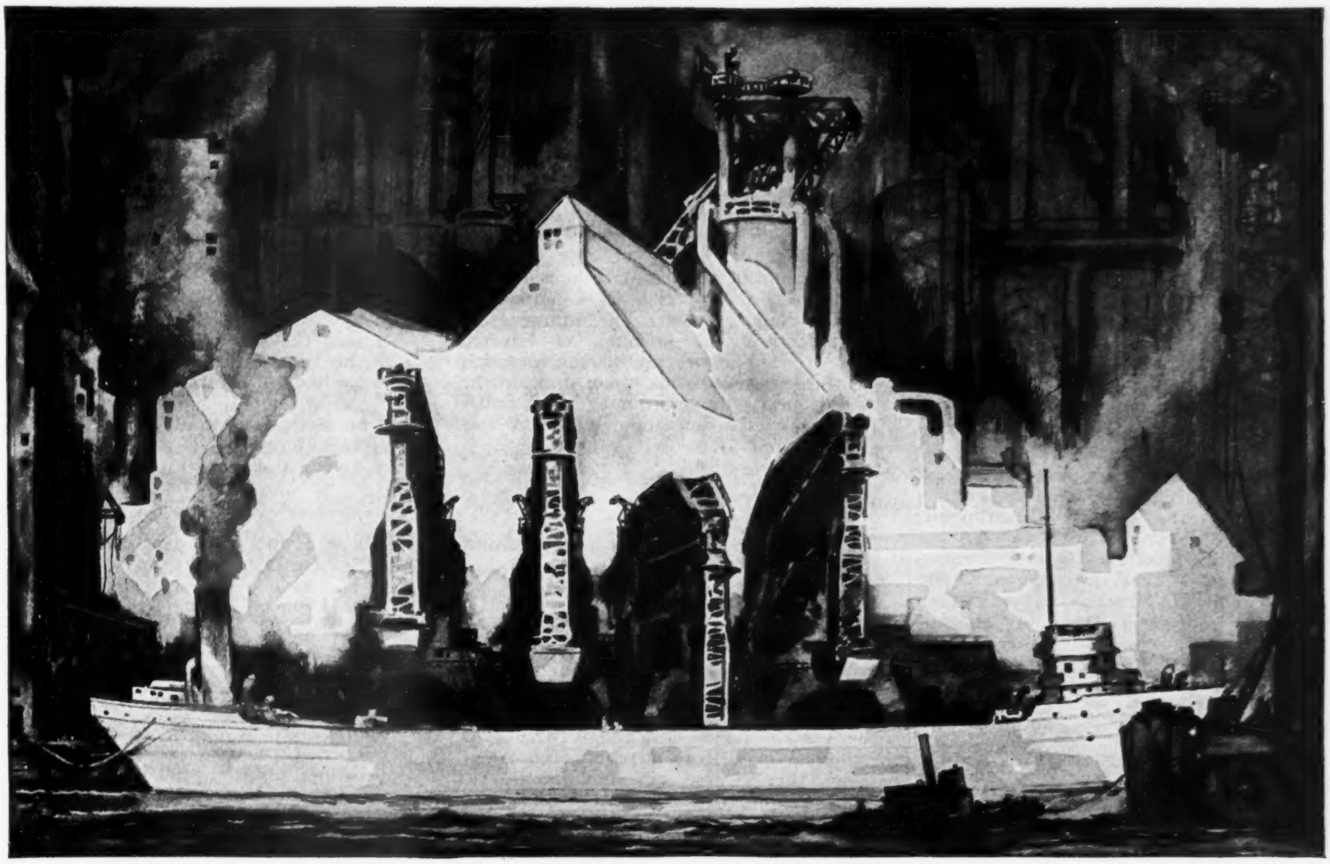

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BOOKS

1—Generating Stations (Second Edition)

By Alfred H. Lovell

438 pages Price $4.50

This is the second edition of Professor Lovell's book which appeared originally in 1930. In the intervening period there have been some developments in the physical aspects of station design, notably in the further application of the mercury-vapor-steam cycle, some new steam turbines of special interest, large scale hydro projects by the Federal Government, further applications of high steam pressures and high voltage for long-distance transmission. These are covered in the present edition, but the major change calling for revision has been in the economic situation which receives extensive attention and has been brought practically up to date by the inclusion of statistics through 1934.

The book does not attempt to go into much technical detail with regard to the design of power stations, but does give an excellent exposition of the economics underlying such design and deals at considerable length with the problems of central station management. Among the subjects covered in this respect are: Principles of Corporate Finance; Cost of Stations; Depreciation and Obsolescence; Load Curves; Power Plant Location; Power Distribution Systems, etc.

2—Corrosion—Causes and Prevention (Second Edition)

By Frank N. Speller

700 pages Price $7.00

This is the second edition of this important treatise, first brought out in 1926. In the intervening period much research has been conducted by various bodies and individuals on this world-wide problem and many reports have been published. The most important of these have been reviewed in preparing the present edition, and the text has been revised to bring it in line with present knowledge on the subject. The form of treatment and classification of factors and types of corrosion followed in the first edition has been continued in this edition. In order to keep the size of the volume within reasonable bounds, the older sections in many cases have been condensed or omitted where better data are now available. Therefore, the first edition may still be used in conjunction with this edition for reference on certain subjects. The mechanism of corrosion is discussed in detail and the practical application of preventative measures is included. Among the more important topics considered are the influence of manufacture, the influence of internal and external factors, methods of corrosion testing, the prevention of corrosion in the atmosphere, underground and under water, deactivation and deaeration and stray-current electrolysis.

3—Manufacture of Seamless Tubes (Ferrous and Non-Ferrous)

By Gilbert Evans

187 pages Price $12.00

Mr. Evans' book is the first on the manufacture of seamless metal tubes that has been published since 1903. Written by a practical man from a practical point of view, this volume will undoubtedly remain a standard work on the subject for many years to come.

The book contains exhaustive and illustrated descriptions of practically every process by which seamless tubes are manufactured. Also, much information is given, based on first hand experience, that will help the tube producer to choose the process best suited to particular conditions.

In addition, the author deals with warehousing, tools, testing, inspection and the general organization of tube plants. The book, therefore, should be indispensable not only to practical operators and metallurgists but also to all manufacturers intending to produce seamless tubes.

4—Air Conditioning and Engineering

332 pages Price $5.00

This exhaustive treatise on the technique of the conditioning and mechanical movement of air has been prepared by the Engineering Staff of the American Blower Corporation and the Canadian Sirocco Company, Ltd. The material has been selected and arranged with a view to providing a ready reference and convenient handbook for engineers having to deal with problems involving air flow and the application of air to ventilating, cooling, etc.

The book is divided into two sections, the first dealing with fundamental engineering information, charts, tables and other data, and the second with equipment, including dimensions and capacity tables.

Among the subjects treated in Section I are heat and steam, the laws of air flow and methods of measurement, ducts and their arrangement, fans for ventilation and for mechanical draft, fan drives, dust collection, humidifying and cooling, process drying and refrigeration.

5—Dissolved Oxygen in Boiler Feedwater

By M. C. Schwartz

46 pages Price 50 cents

This pamphlet issued by the Louisiana State University Press, Baton Rouge, La., describes a refined technique for the determination of dissolved oxygen as developed at the Louisiana Steam Generating Corporation's laboratory. The method is recommended for oxygen concentrates under 0.5 ml per liter. Section II of this 46-page pamphlet is devoted to a selected bibliography on the determination of dissolved oxygen in water.

6—Introduction of Mechanics and Heat

By N. H. Frank

332 pages Price $3.00

The subject of mechanics and heat at an intermediate level is discussed under the following heads: fundamental definitions, linear kinematics of a mass point, plane kinematics of a mass point, Newton's Laws, statics of a particle, linear and plane dynamics, work and energy, potential energy, special dynamics of a mass point, dynamics of a system of particles, statics of rigid bodies, plane dynamics of rigid bodies, special rigid motions, planetary motion, gravitation, hydrostatics, fluid dynamics, static elasticity, dynamics of elasticity, acoustics, temperature and thermometry, the first law of thermodynamics, heat conduction, thermodynamics and kinetic theory of an ideal gas, the properties of real gases and the second law of thermodynamics.

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EDITORIAL

Viewing the Utility Situation

Among the factors that at present stand out in the utility situation are:

1. A test of the constitutionality of the recent holding company legislation has already been started in the U. S. District Court of Maryland. The trial, decision, appeal and final decision by the U. S. Supreme Court are likely to take many months—too long to have much bearing on immediate engineering requirements.
2. The S.E.C. and the utility representatives have held preliminary conferences pertaining to the administration of the Act, without prejudicing the latter's rights subject to court decision. This step may ease the financing of new work in certain instances.
3. Several companies have already severed physical power connections across state lines in order to be free from certain restrictions imposed by this legislation. If this procedure is widely carried out it will affect the adequacy of existing reserve capacity.
4. The utility load continues to forge ahead. It is now about thirteen per cent over that of a year ago, has practically equalled the all-time record peak and since early summer has been well above the corresponding load for the banner year of 1929. Moreover, this increase is no longer attributable largely to domestic load, for industrial load has been fast coming back. Diversity and peaks, of course, tell more than total kilowatt-hours in judging capacity to meet the load.
5. Here and there, engineering studies that have been dormant for some time are beginning to take the form of active projects.

It appears from the foregoing that resumption of normal activity in those branches of the capital goods industries affected by central station buying may be closer at hand than was anticipated six months ago.

It is reasonably certain that few, if any, new stations will be built in the near future. Present indications point to replacement of existing equipment or superposition of high-pressure units on existing low-pressure equipment. Yet one well-known utility engineer contends that superposition is only a passing expedient, comparable to the low-pressure turbine coupled to the exhaust of a reciprocating engine—an arrangement that was popular about twenty-five years ago. But the low-pressure turbine, by adding capacity and increasing economy, performed a very useful purpose, just as superposition is likely to accomplish in the immediate future, regardless of ultimate trends.

Another well-known utility engineer and executive is reported as strongly adverse to concentration of power in large units and large stations. However, most of the present utility construction and pending proposals involve large units, some larger than heretofore attempted.

Such diversity in opinion is wholesome and promotes progress which after all is what is needed in a field that has been almost devoid of construction for four years.

Cross-Drum Boiler Circulation

This year's Prime Movers Report on "Boilers, Economizers, Air Heaters and Piping" devotes considerable space to the difficulties which operating companies have encountered through poor circulation in the upper tubes of boilers of the cross-drum type. These difficulties are inherent in the type, regardless of the make, where the boilers are operated at high pressures and high rates of heat transfer; nor are they confined entirely to the utility field. The region of water *starvation* lies in the upper part of the upper rows of tubes, near the uptake header and manifests itself in excessive metal temperatures and occasional corrosion. As the load increases these dry areas extend downward.

Various remedial measures have been applied with varying degrees of success. These have included rebaffling; the insertion of ferrules in the connecting nipples which join the upper and lower tube sections; additional circulating tubes; and the insertion of cores in the tubes that showed faulty circulation. The last-mentioned measure appears to have been successful in several instances.

The problem is receiving careful study by both the operating companies and the manufacturers' engineers. Measurements of metal temperatures by thermocouples permits the location and extent of dry areas at different rates of driving to be determined and other means are being employed to indicate reverse circulation. Investigations have thrown much light on the subject, and indicate that our previous conceptions of circulation within this type of boiler, based on operation at lower pressures and lower heat transfer rates, will have to be revised.

It is significant that the report mentions no circulation difficulties with boilers of the bent-tube type.

Contribution to Piping Knowledge

The investigations of Lieutenant Dennison at the U. S. Naval Engineering Experiment Station on the strength and flexibility of corrugated and creased pipe bend forms a valuable addition to the earlier studies by Bantlin and Karman abroad, Professor Hovgaard at M. I. T. and the more recent investigation by Messrs. Cope and Wert of the Detroit Edison Company which have been reported in papers before the A.S.M.E.

Employment of higher steam temperatures increases the expansion which must be absorbed within the piping system and also compels the designer to take cognizance of creep. In naval practice where space is usually very limited expansion bends must be a minimum; hence the interest in corrugated and creased pipe which may be bent to a much smaller radius than is practicable for smooth-walled pipe and which exert lower forces against the anchor points. While naval pressures and temperatures are lower than those employed in many land stations, the results of this latest investigation contains much that can be applied to stationary practice.

COAL SAMPLING METHODS of The Detroit Edison Company

By A. W. THORSON
The Detroit Edison Company

The automatic mechanical sampling devices employed at the plants of this company are of the intermittent type, one form being used with bucket conveyors and the other with belt conveyors. With the former a $^1/_{10}$ of one per cent sample (35-lb increment) is taken every $7^1/_2$ min and with the latter a 14-lb increment is taken once per minute. A special design of riffle box has been developed to eliminate clogging with wet coal.

THE vital steps in securing data on the coal which a power house consumes are:

1. Collecting a sample
2. Reducing this gross sample to laboratory size
3. Analyzing the laboratory sample

To illustrate the full importance of performing these three steps correctly, consider the case of a large central station burning 1000 tons of coal per day. The heating value of the coal is usually determined weekly from a one-gram sample. Since there are almost a million grams in a ton, nearly seven billion grams are consumed in the week. Therefore, it is imperative that the greatest precaution be used in choosing one portion in seven billion on which to base computations of efficiency. The problem is further complicated because coal is a heterogeneous material to which ordinary sampling theories do not apply. Determination of sample size, therefore, has been done largely by empirical methods.

Factors Affecting Size of Sample

Two major factors affecting the size of sample are moisture content and size distribution of the coal. Size distribution affects the size of sample, for obviously the sample should be larger as the size of the largest pieces of coal increases. In addition, the proportion of various sizes will have its influence, because pieces of the same size may have different properties, variance of which depends on the seam from which the coal was obtained. Hence, it is necessary that the sample be of such quantity that its sieve analysis, as determined by A.S.T.M. Designation D311-30, "Sieve Analysis for Crushed Bituminous Coal," be identical to that of the whole lot of coal.

Moisture content also influences sampling by affecting the size distribution. Excessive moisture may increase the proportion of fines adhering to large pieces so that a sample having moisture content higher than the average will not have average size distribution. Thus a sample, the correct size of which has been determined for one moisture value by a sieve analysis, may

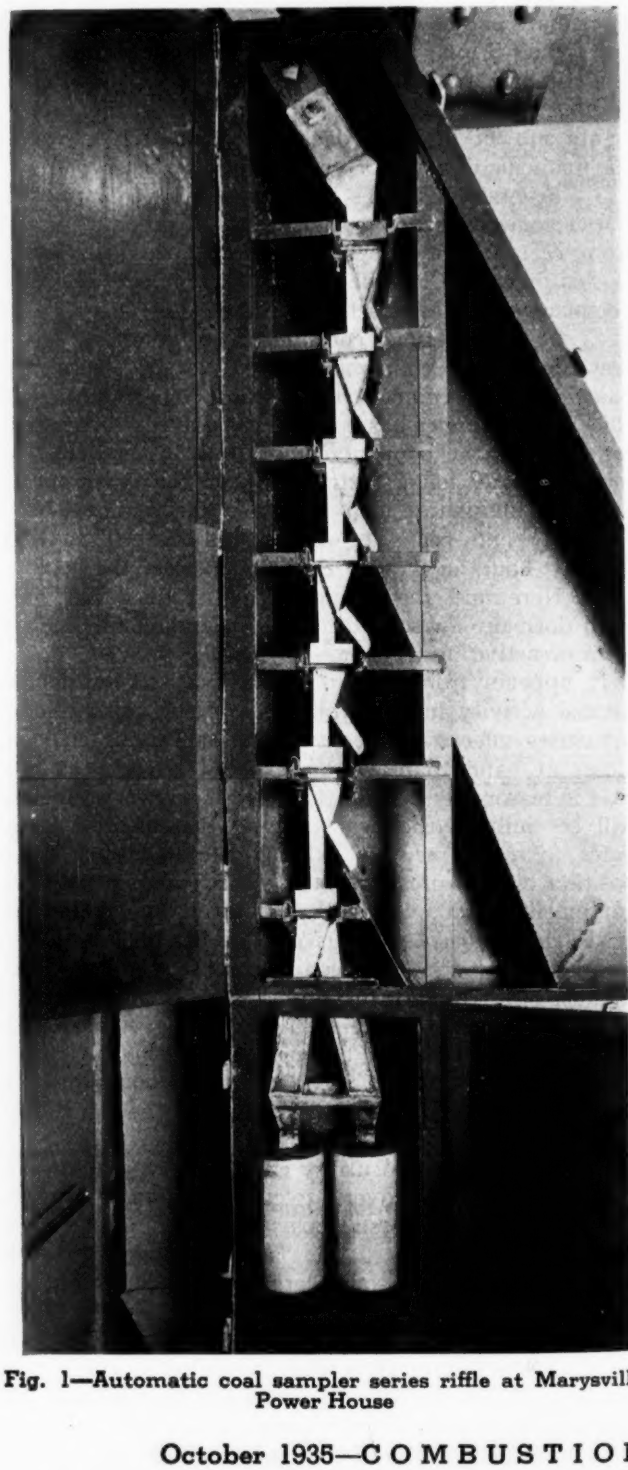


Fig. 1—Automatic coal sampler series riffle at Marysville Power House

not be the proper size for another moisture value. It is, therefore, necessary to consider both size distribution and moisture content in deciding on the weight of sample to be collected.

Correct Size of Hand Sample

The A.S.T.M. Standard Designation D21-16, "Method for Hand-Sampling of Coal," prescribes a sample 0.1 of one per cent by weight of the coal it represents. Where small trial lots of coal are tested, it is necessary to take this size of sample. However, a different condition arises when a large quantity of coal from one source is being burned continuously. Crushing and thorough mixing in the coal preparation equipment reduces the required size of sample. Tests performed by The Detroit Edison Company showed that for such conditions, a sample of 0.01 of one per cent, carefully obtained from the stoker hoppers, gave analyses equivalent to those obtained from a 0.1 of one per cent sample. As a result, hand samples at the power houses of the Company were reduced to a minimum of 0.01 of one per cent of the coal burned.

Correct Size of Mechanical Sample

With the advent of mechanical sampling methods, however, a new problem arises as to the correct size of sample to be collected. Two methods present themselves; (1) continuous sampling of a portion of the cross-section of the coal stream, and (2) intermittent sampling of the entire cross-section. The choice of the second method is inevitable unless the coal be so finely crushed that there is no variation in size or properties over the cross-section of the stream. This condition is not encountered even in a pulverized coal stream.

A gross sample collected by the intermittent method must consist of a number of increments of equal weight taken at equal increments of time. Thus, decision as to the proper weight of sample resolves into determination of the size of the increments and of the frequency with which they are collected. It is self-evident that if an increment is to represent anything, it must be large enough so that its size distribution equals that of the lot of coal. The minimum weight of increment can therefore be determined by experiment. The frequency of sampling, however, may not be so easily decided upon. If the properties of a given size of pieces vary, the greater the frequency, the more accurate the sample. As one cannot take an infinite number of increments, the minimum allowable number must be determined experimentally for each case, depending on the kind of coal, its size and the desired degree of accuracy.

The remainder of this article is devoted to a description of the equipment used in the mechanical sampling of coal in the power houses of The Detroit Edison Company.

Automatic Coal Samplers

In developing mechanical sampling devices, the necessity was seen for two types because both belt and bucket-conveyors are in use in the power houses.

Bucket Conveyor Type

The Peck type carrier, from which it was decided to obtain samples, consists of a chain of buckets of rectangular, horizontal section and parabolic, vertical section. Each bucket is fastened to the chain by pivots

Fig. 2—Coal sampler bucket, vertical section reduced

at the top center of the two ends, and is dumped by cams on the bottom which strike their followers over the hopper that feeds the coal bunkers.

To obtain a sample increment, it was necessary merely to replace the standard cams on a bucket with special ones and install trippers over the sample collecting hopper to strike these cams. The sampling hopper could thus be located at any accessible point along the horizontal run of the conveyor. Since there are approximately 170 buckets to the chain, one sample bucket would give 0.6 of one per cent gross sample which was much too large. Consequently, a false bottom was placed in the sampler bucket to reduce its capacity. This was accomplished by reducing the vertical section in one case, as shown in Fig. 2, and the horizontal section in another, as in Fig. 3, with equally good results. By determining the sieve analysis of the samples from

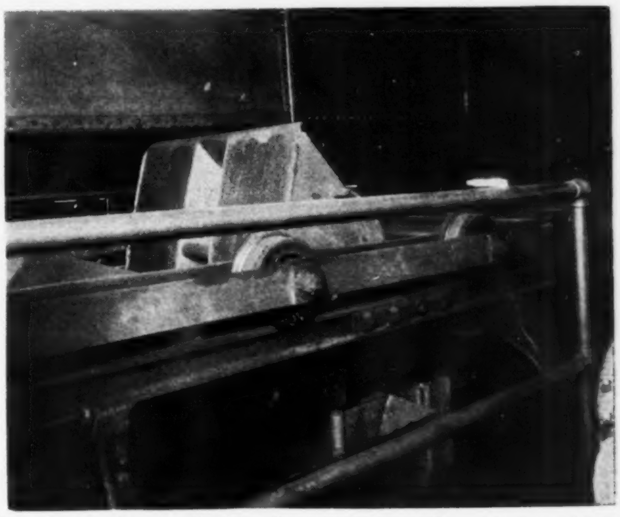


Fig. 3—Coal sampler bucket, horizontal section reduced

various bucket capacities and comparing each with a large sample containing approximately the same moisture content, the minimum permissible size of sample increment was found to be about 25 lb. One sampler bucket per chain, giving a sample of about 0.1 of one per cent by weight of coal fed to the bunkers, was found to give a representative gross sample where a continuous supply from the same sources was being consumed. However, where test shipments of a few cars of coal are sampled, it has been found advisable to collect three increments per chain, or about 0.3 of one per cent, in order to insure a representative sample of the lot. This was solely to increase the frequency and not the sample weight. At the power house where the test coals are received, extra riffles are used in the series to reduce the larger gross sample to laboratory size for normal coal, and these are by-passed when sampling test coal.

This sampling device is shown schematically in Fig. 4. Coal dumped into the sample hopper is piped to the rotary crusher where it is reduced in size to $^1/_4$ in. The sampling slot of this crusher retains about 5 per cent, which portion is piped to the series riffle box, shown in Figs. 1 and 5. Lack of symmetry about the vertical axis of these riffles is the result of experience with wet coal. The conventional design clogged badly when coal with excessive moisture was encountered and the change in design presents a nearly vertical path to the flow of coal and clogging is thereby eliminated. The last riffle in the series of standard design, divides the retained sample and deposits the two portions in two aluminum containers with dust-tight covers. The number of riffles and size of containers are regulated so that the cans will hold one day's collection of sample. At the end of the day the two cans are removed, and metal covers with expanding rubber gaskets are used to seal the samples, allowing no air space above the coal. One of each daily pair of samples is sent to the laboratory and the other is held in reserve until the analyses have been received. The samples thus obtained are used for all determinations including moisture.

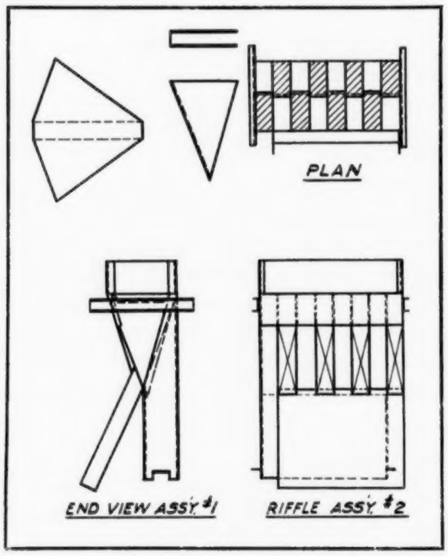


Fig. 5—Details of riffle

This device is entirely automatic in operation as the sample crusher motor is interlocked electrically with the conveyor motors. The only attention normally required is the removal and replacement of the sample cans at the end of the day, except in cases where rubbish and extremely wet coal may at times plug the system.

Belt Conveyor Type

A relatively simple device was worked out for the bucket conveyor, but the design of that for the belt conveyor is more complicated. It is, of course, desirable to locate the sampler where the cross-section of the coal stream is a minimum in order to reduce the size of sampling device required and the weight of sample increment. In this particular installation, no location ahead of the belt conveyor was available. It was deemed inadvisable to sample from the belt itself, for any device for dipping the coal from the belt must scrape the belt to

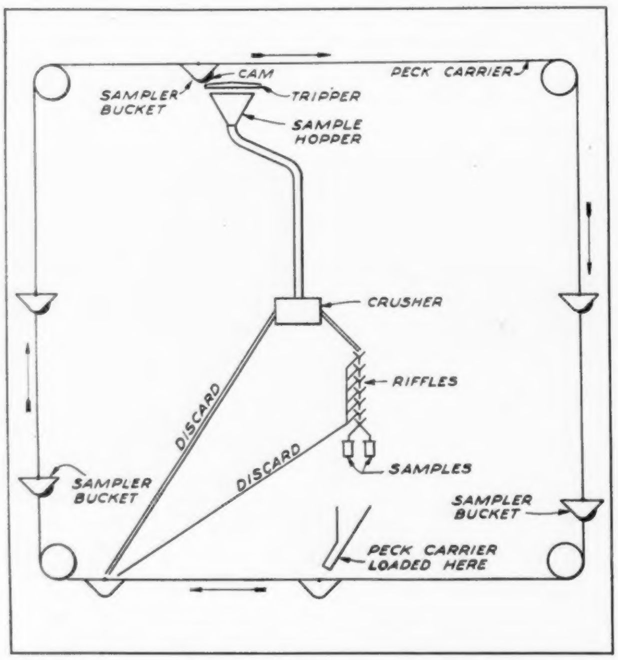


Fig. 4—Diagram of automatic sampler for bucket conveyor

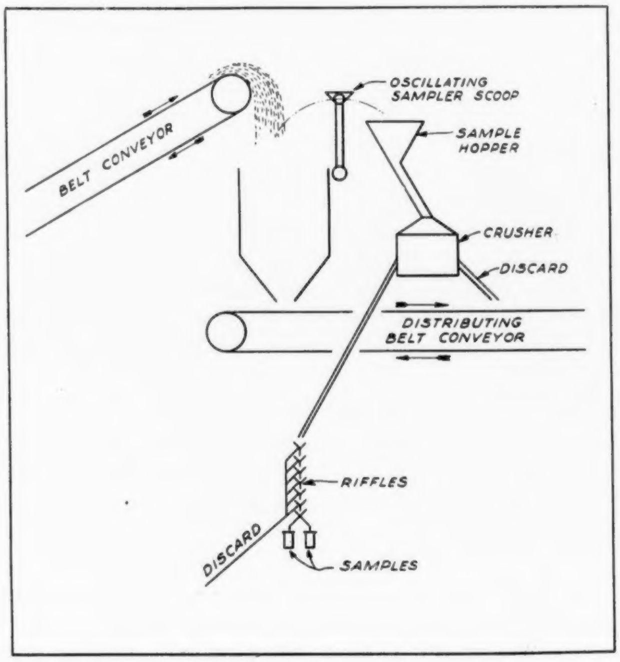


Fig. 6—Diagram of automatic sampler for belt conveyor

Fig. 7—View of automatic sampler, coal crusher and riffle box in Trenton Channel Power House

obtain the entire cross-section, thereby damaging it. The logical location, therefore, was just under the head pulley where the coal stream leaves the belt and dumps into a hopper.

The sampling device developed for this location is shown on the schematic diagram, Fig. 6. An oscillating sampler scoop, details of which are shown in Fig. 8, traverses the coal stream with suitable speed to fill completely just as it leaves the stream of falling coal. It returns and dumps into the sample hopper whence crushing and riffling are done as with the bucket conveyor samplers. One of these crusher and riffle installations is shown in Fig. 7.

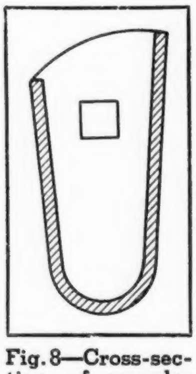


Fig. 8—Cross-section of sampler scoop for belt conveyors

The scoop design shown is the result of a number of experiments to determine the proper size and shape which would give the correct sieve analysis and which would collect a sample representing the entire cross-section of the coal stream. The capacity of this scoop is about 14 lb of $1^1/_2$-in. crushed coal.

The sample scoop is driven by a one-horsepower, constant-speed motor through an adjustable-speed transmission and worm-gear reducer to a cam. This cam is so shaped that one revolution moves the scoop through one complete cycle of operation. The speed of the cam is controlled by adjustment of the transmission device. At present, the cam is rotating about one revolution per minute, causing the scoop to make an oscillation in approximately one-half minute and remain over the sample hopper the remainder of the period. The axial position of the scoop is controlled by a crank bolted to each end of the scoop which engages stationary guides. These guides cause the scoop to pass through the coal stream in an upright position and then to dump into the sample hopper. These details are shown in Fig. 9.

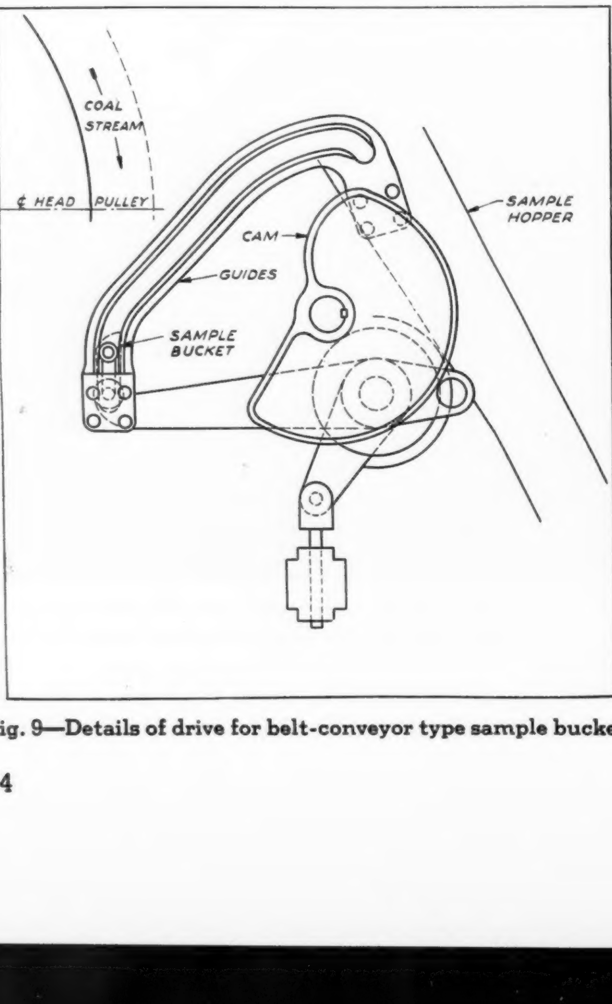


Fig. 9—Details of drive for belt-conveyor type sample bucket

This device has operated satisfactorily, with the exception that extremely high moisture content caused the coal to stick in the scoop. The difficulty was remedied by installing two small hammers which strike the scoop just after it dumps. These hammers are pivoted from a rod fixed above the dumping position of the scoop and are held in suspension by a spring. The scoop mechanism trips this spring as it reaches the end of its travel.

The gross sample obtained by this device is composed of 14-lb increments taken about one minute apart. This constitutes approximately 0.1 of one per cent of the coal passing the conveyor. The weight of increment and sampling frequency were determined by the same methods as were employed for the bucket-conveyor samples, and give about the same size of gross sample, as would be expected.

Summary

The correct weight of a coal sample depends principally on the size of the largest pieces, the size distribution or sieve analysis, and the moisture content of the coal to be sampled.

There are two common methods of collecting a sample: (1) continuous deflection of a portion of the coal stream, and (2) intermittent deflection of the entire coal stream. The latter is the more desirable method.

In sampling by the intermittent method, two quantities must be decided upon: first, the weight of each increment, which must be sufficient to have the same sieve analysis as the lot of coal being sampled, and second, the frequency of collecting increments, which must be sufficient to account for variations in the properties of any one size of pieces.

Using these fundamentals as a guide, two automatic sampling devices described in detail were developed—one for bucket conveyors, and one for belt conveyors.

The correct size of sample for the bucket conveyor was found by experiment to be about 0.1 of one per cent of the coal consisting of 35-lb increments of $1^1/_2$ in. crushed coal taken every 7.5 min. For the belt-conveyor type also, 0.1 of one per cent was decided upon, consisting of 14-lb increments of $1^1/_2$ in. crushed coal taken once per minute.

T. I. Phillips has been appointed General Works Manager of the Westinghouse Electric & Manufacturing Company. He has been associated with that company, in various capacities, since 1915 and succeeds C. H. Champlain who has been forced by prolonged illness to relinquish his duties.

W. P. Saunier, formerly with W. S. Barstow & Co., is now with the Cambridge Electric Light Company, Cambridge, Mass.

David J. Champion, President of the Champion Rivet Company, Cleveland, O., and well known in the boiler manufacturing field, died on Sept. 10, 1935 at the age of seventy-five.

DISTRICT STEAM HEATING—II

By ROGER D. DeWOLF
Consulting Engineer,
Rochester, N. Y.

In the September issue, the writer discussed the general background and outlined the unfavorable conditions under which district steam heating labored up to 1920, and the betterment of those conditions between 1920 and 1925. Data and curves were presented showing the very satisfactory growth in sales and earnings of a number of the companies, several showing increasing sales right through the depression period. The present article shows further the beneficial effects of changes in sales policy and improvements in operating practice; rates and load factor are discussed as well as by-product electricity. The economics of the physical aspects of steam generation and distribution will be taken up in the final article next month.

DURING 1923–24–25, there was a growing conviction that steam heating was in reality steam service; that an aggressive sales campaign would convince its customers that it possessed many real advantages; and that it paid to give the customer real service and cooperation along with the steam.

Sales efforts have been gradually familiarizing customers with the economic use and great convenience of steam service. Among the advantages stressed are the betterment of public health through reduction of smoke; a reduction in the amount of dirt and dust which open truck loads of coal and ashes spread upon the streets; a reduction of fire hazard; the value of basements for additional sales and storage space, cafeterias, barber shops, etc.; and the values, tangible and intangible, which such service gives over and above the mere cost of coal required for private steam generation. Experienced service engineers and automatic control devices have, in some cases, effected reductions in customer's steam consumption of as much as 50 per cent.

These efforts resulted in very greatly improving the status of many of the companies. The results for three companies averaged for the years 1924–25 and for 1928–29, are shown in Table IX.

TABLE IX

Average for Three Companies for 1924–25 and for 1928–29

Years	Steam Sold 1000 M Lb	Avg. Gross Income	Avg. Net Income	Avg. Op. Ratio	Avg. Rate
1924–25	3405	$3,277,000	$1,363,000	0.58	$0.96
1928–29	5094	$4,491,000	$2,500,000	0.45	0.88
Increase	57.5%	37.5%	83.5%		
Decrease	....	....	....	29%	8.4%
Average of the 4 Years					
1924–25 } 1928–29 }	4250	$3,884,000	$1,932,000	50%	$0.91

During this period, the ten companies included in Group 3 of Table V[1], showed the following growth (Table IX-A):

Table IX-A

Years	Steam Sold 1000 M Lb	Avg. Gross Income
1924–25	6975.7	$5,840,000
1928–29	9471.8	$7,842,200
Increase	35.8%	34.1%

The 1924–25 period was a long step forward from 1920, and shows the beneficial effects of the change in policy, in sales effort and in operating practice during that four-year period. The investment can be approximately calculated by using a figure of $4.00 investment per $1.00 of gross income.

Major additions to plant and distribution systems were made during the years 1924–25 to 1928–29: as these additions were based upon the rate of growth established prior to and during these years, there was unsold capacity available in 1929, with the result that the investment ratio became approximately 5 to 1 rather than 4 to 1. The average investment and average net income expressed as a per cent return on the investment is shown in Table X.

TABLE X

Years	$ Invest. per $ Gross Inc.	Total Investment	Net Income as % Return
1924–25	$4.00	13,150,000	10.4
1928–29	5.00	22,500,000	11.1
Increase		71%	
Average of the 4 Years			
1924–25 } 1928–29 }	5.00	19,500,000	10.1

During this period, the investment increased 71 per cent, or $9,350,000., and the net income $1,137,000. Despite the fact that this large increase in investment resulted in a considerable amount of unsold capacity in 1928–29, the return on this additional investment amounted to practically 12 per cent, and if this surplus capacity had been sold, so that the investment ratio had been brought down even to $4.50 per $1.00 of gross income, the return would have been 14.8 per cent as shown below:

Gross income would have been $5,000,000
Net income would have been $2,750,000

The averages for these three companies can be taken as being a fairly representative cross-section of steam service to various types of customers, from small, low load factor space-heating customers, to large industrial plants using steam for process work throughout the year, in addition to space heating, with a yearly load factor of 40 to 50 per cent.

With the figures of 1924–25 and 1928–29 as a basis, Table XI gives the financial results for such an average

[1] See the first article of this series, COMBUSTION, September 1935.

company, set up on the following fundamental data:

Period	Investment Ratio	Income per $1.00 Invested
First 5 yr.	$5.00 per $1.00	$0.20
Next 10 yr.	4.75 per 1.00	0.21
Last 10 yr.	4.50 per 1.00	0.22

Operating ratio 50 per cent
Average rate per M lb $0.90

Total investment $19,500,000 financed on the basis of approximately 60 per cent first mortage 5 per cent sinking-fund 25-year bonds, callable at 102, bond discount $500,000 and 40 per cent common stock; the bond discount to be written off in 25 years.

5 per cent sinking fund bonds issued......$12,000,000
Common stock..........................$ 8,000,000

Period	Gross Income	Net Income
First 5 yr.	$3,900,000	$1,950,000
Next 10 yr.	4,095,000	2,047,500
Last 10 yr.	4,290,000	2,145,000

Taxes taken at 12¢ per $1.00 gross income.

Accumulated surplus after dividends and in excess of $500,000 is to be reinvested to yield 4 per cent, and this income added to surplus. Total payments for sinking fund, bond discount and taxes are as follows:

	First 5 yr.	Next 10 yr.	Last 10 yr.
Sinking fund	$489,600	$489,600	$489,600
Bond discount	20,000	20,000	20,000
Taxes	468,000	491,400	514,800
	$977,600	$1,001,000	$1,024,400
Net operating income	1,950,000	2,047,500	2,145,000
Net before interest	$972,400	$1,046,500	$1,120,600

TABLE XI

Earnings Based on Averages for 1924–25–28–29
Figures Are in $1000

Year	Bonds Outstand.	Bond Int.	Net to Stock	Dividend $	Dividend %	Accum. Surplus	4% Int. on Acc. Surp. over $500,000
1	$12,000	$600	$372.4	$320	4	$52.4	
2	11,520	576	396.4	320	4	128.8	
3	11,040	552	420.4	400	5	149.2	
4	10,560	528	444.4	400	5	193.6	
5	10,080	504	468.4	400	5	262.0	
6	9,600	480	566.5	480	6	348.5	
7	9,120	456	590.5	480	6	459.0	
8	8,640	432	614.5	480	6	593.5	
9	8,160	408	638.5	480	6	755.7	3.7
10	7,680	384	662.5	480	6	948.4	10.2
11	7,200	360	686.5	480	6	1172.7	17.8
12	6,720	336	710.5	480	6	1430.0	26.8
13	6,240	312	734.5	480	6	1721.6	37.1
14	5,760	288	758.5	480	6	2048.9	48.8
15	5,280	264	782.5	480	6	2413.2	61.8
16	4,800	240	880.6	560	7	2810.3	76.5
17	4,320	216	904.6	560	7	3247.4	92.5
18	3,840	192	928.6	560	7	3725.8	109.8
19	3,360	168	953.6	560	7	4247.4	129.0
20	2,880	144	976.6	560	7	4814.0	150.0
21	2,400	120	1,000.6	640	8	5347.4	172.8
22	1,920	96	1,024.6	640	8	5925.5	193.5
23	1,440	72	1,048.6	640	8	6550.6	216.5
24	960	48	1,072.6	640	8	7224.7	241.5
25	480	24	1,096.6	640	8	7950.3	269.0
			$18,733.0	$12,640	6.32 Avg.		$1857.3

In the calculation of Table XI note the following points in which it has been conservatively estimated.

1. The lowest investment ratio used was $4.50 per $1.00 gross income, whereas experience has shown that where the construction costs are not abnormally high, this ratio becomes more nearly 4 to 1 when the system becomes fully loaded. This means that the income per dollar invested in the last 10 years would be nearer $0.25 than $0.22, and the gross income increased some $500,000 per year.

2. A 2 per cent premium has been set up for calling the bonds; with sinking-fund bonds issued serially, this would be unnecessary.

3. For bonds of this type, $500,000 bond discount for a $12,000,000 issue is excessive: $2^1/_2$ per cent discount should be entirely sufficient.

4. The accumulated surplus is reinvested in other securities paying only 4 per cent: much of this money can generally be reinvested in extensions to the property and earn a much higher return.

Even with these conservative factors, at the end of the 25-year period the surplus is sufficient to pay a 100 per cent common stock dividend in cash or securities, after paying an average of $6^1/_3$ per cent regular dividends throughout the period, leaving the property in well maintained and going condition in the hands of the common stockholders, free and clear of incumbrances. Just how much more than this the common stockholder would be permitted to earn, is a question which would probably have to be settled by the Public Service Commissions; the additional earning capacity present in this analysis would, perhaps, be used to better advantage in making rate reductions.

Rates and Load Factor

By 1924 the average rate per 1000 lb at which steam was being sold was pretty well stabilized for the individual companies, although the rate varied over quite a wide range between different companies. A number of factors account for this variation in rates between different companies; those principally affecting the average rate at which the entire output of a company is sold are:

a. The Price of Fuel.—This is the largest single item in the operating cost.

b. The Annual Load Factor.—With an investment of approximately $4.50 per $1.00 of gross business, the fixed charges (made up of depreciation, taxes, interest and dividends to the stockholder) total up to an amount approximately equal to the total operating costs. The higher the load factor, the lower these charges become per 1000 lb of steam sold, since more steam is sold in the course of a year from a given total investment.

c. Average Yearly Consumption per Customer.—All modern rate systems, of whatever type, are designed to give automatically the large consumer a lower rate per 1000 lb than the small consumer; hence, the larger the customer the lower the rate, with generally an improvement in the load factor and net income.

The N.D.H.A. proceedings show that for 1929 the rates of thirty-nine different companies varied between $0.52 and $1.15 per 1000 lb with coal prices varying between $2.26 per ton for coal running 13,050 Btu to $7.60 per ton for coal running 13,000 Btu per lb; and with annual load factors varying between 16 and 42 per cent; and with the average yearly consumption per customer varying between 108,000 lb and 9,235,000 lb.

With these three factors varying over such wide limits, it is quite natural that the average rate should vary widely between companies. In a few cases, certain local conditions, sometimes of a political nature and sometimes due to a low price basis having become too firmly established, are the controlling factors.

The rates of these 39 different companies fall into the following classification:

Avg. Rate	No. of Companies	Total Sales 1000 (M) Lb
$1.00 and above	5	3,584,494
0.90 to $1.00	10	12,034,579
0.80 to 0.90	9	4,478,530
0.70 to 0.80	8	2,696,566
0.59 to 0.70	4	4,252,684
0.50 to 0.59	3	1,893,833
	39	28,940,686

Of the total steam sold, 54 per cent is sold at an average rate of $0.90 per M lb or higher, and only 4.1 per cent is sold at less than $0.59.

A very large proportion of the companies use the so-called block rate, in which the rate for the first 100,000 lb used per month is high, for the next 100,000 lb somewhat lower and so on by successive steps. About 1926 a few companies put in an "optional demand rate," and the outstanding development in the rate situation for the past ten years has been the growth in the adoption of this form of rate.

Because the demand rate requires the installation of quite expensive metering equipment, it is not adaptable to the small consumer whose yearly bill does not warrant the heavy investment in metering equipment with its relatively high maintenance and operating cost. For the large consumer, however, especially where steam is used for other purposes than space heating, its use is becoming more general. The demand rate is based upon the fact that the customer who uses a large amount of steam in the course of a year from, say, 100 hp of boiler capacity and distribution capacity, reduces the fixed charges per 1000 lb of steam, as outlined in paragraph *b* above, and hence is entitled to a lower rate than a customer who requires the same maximum demand in boiler horsepower and distribution capacity but uses only half as much steam in the course of a year.

For example, two customers may each have a maximum demand of 6000 lb of steam per hour, each requiring the same investment in steam generating capacity, distribution line capacity, etc. One customer, with a load factor of 50 per cent will use 15,840,000 lb of steam per year, while the other with a load factor of only 25 per cent will use only 7,920,000 lb per year; the fixed charges on the investment per 1000 lb of steam will be only half as much for the 50 per cent load factor customer as for the 25 per cent load factor customer. The demand rate is designed so as to give automatically the high load factor customer the lower rate.

This rate, from the viewpoint of the utility, also has the advantage of enabling the utility to compete more effectively with the isolated plant, and to get business which, under the block rate system, it would be unable to secure. It also makes possible taking on summer business at a rate attractive to both the user and the utility.

Reports Show Improved Load Factor

There has been a striking improvement in the annual load factors of the companies reporting to the N.D.H.A. In 1925 only 3 of the twenty-six companies reporting, or 11.5 per cent, showed a load factor higher than 27 per cent, the highest reported being 31.2 per cent. In 1929, twenty-five out of thirty-nine companies reporting, or 64 per cent, showed a load factor higher than 27 per cent, the highest reported being 42 per cent. As the load factor has increased, there has been an accompanying lowering of average rates.

Under this rate system, the demand is usually determined during December, January and February and it becomes worth while for the customer so to operate his heating and other steam using equipment as to keep the maximum amount used in any one hour during these months as low as possible. The improvement in load factor as noted is probably due as much to existing customers having improved their load factor, as to new high load factor customers having been taken on.

Relation of Electric Service and Steam Service

A study of the relative magnitude of electric service and steam service for customers of different sizes and types brings out some very interesting comparisons.

The electric utilities have found, as a rule, that any department store, hotel or other prospective customer using around eight to ten million pounds of steam per year or more, and with an average electric demand, is in the competitive class; that is, such a customer may be able to generate his own steam and electricity at a cost comparable to the price at which the utility can supply it. It has also been found that if the customer has to generate his own steam, the additional cost of generating his current compares favorably with the utilities price, unless the electric load is greater than is usually the case, or the summer electric demand rather high.

A study of some forty odd customers in the competitive class, purchasing both steam and electricity, including department stores, hotels, office buildings, factories and other miscellaneous users, located in several of our larger cities, gives the following averages:

Total income, steam and electricity	$1,750,000
Per cent of total from steam sales	55.5%
Per cent of total from electric sales	44.5%
Average rate per 1000 lb steam	91.5¢
Maximum rate per 1000 lb steam	112.0¢
Minimum rate per 1000 lb steam	55.0¢
Average rate per kwh	2.05¢
Maximum rate per kwh	6.78¢
Minimum rate per kwh	1.26¢

The maximum and minimum figures do not mean much unless the load factor, type of rate used and other data are considered in connection with the rate. They are given mainly to indicate the wide range of conditions of operation of the customers' equipment, within which combined sales of electricity and steam can be made.

In the past, the opinion has been rather widely held that if the customer's cost of steam amounted to 60 per cent or more of his total combined steam and electric cost, it was very difficult for the utility to compete successfully. The following figures show quite conclusively that such is not the case:

For 3 customers steam is 80% or more of the total
For 7 customers steam is 70 to 80% of the total
For 11 customers steam is 60 to 70% of the total
For 10 customers steam is 50 to 60% of the total
For 10 customers steam is 40 to 50% of the total
For 5 customers steam is 30 to 40% of the total

The twenty-one customers whose steam cost is 60 per cent or more of the combined steam and electric cost, use approximately 44 per cent of the total steam sold to all of the customers. In other words, the utilities which were in a position to render both steam and electric service, were actually selling almost half of their large customer sales to customers generally considered as being very difficult prospects to sell either steam or electricity.

Equally important is the fact that the steam sales formed 55 per cent of the total combined sales of both steam and electricity, and that most of the electricity could not have been sold unless steam had been available to sell along with it. These facts indicate that combination sales are just as necessary to day as they were twenty-five years ago, when the utilities were forced in-

to the steam business. The returns on the steam business today are, however, far more satisfactory, and the steam business is standing on its own feet to a greater extent than ever before. The downward trend of electric rates in the past twenty years, has made it possible to raise steam rates to a point commensurate with the value of the steam service, and yet leave the charge for combined steam and electric service well within the limits of the competitive field.

Generation of By-Product Electricity

There appears to have been a trend toward more generation of by-product electricity in recent years, together with the distribution of both high- and low-pressure steam. The N.D.H.A. reports show that in 1929 there were fourteen companies distributing both high- and low-pressure steam, with five companies distributing only low-pressure steam. The results for these companies were as follows:

No. Co.	M Lb Steam Sold Ex. St.	M Lb Steam Sold Live St.	1000 Kwh Generated	Kw Capacity	Steam per Kwh
14	4,629,502	4,168,114	147,520	68,675	31.4
5	738,672	0	20,982	26,350	35
19	5,368,174	4,168,114	168,502	95,025	32

The possibilities of this by-product generation when using modern equipment now available, are much greater than shown above. The average water rate of 32 lb per kwh can be reduced at least one-third when pressures of the order of 400 lb are used with a moderate degree of superheat.

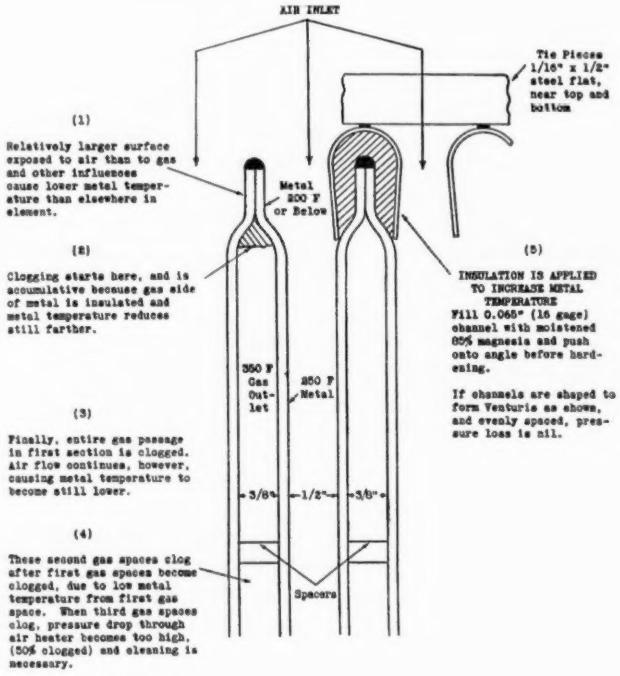
These modern developments in power plant equipment and in the distribution system, their effect upon the general problem of selling either steam alone or combined steam and electricity, will be discussed in the next and concluding article.

Don S. Walker Becomes District Sales Manager

Don S. Walker has lately been appointed District Manager in the Philadelphia office of Combustion Engineering Company Inc. In this capacity he will be responsible for sales in both the Philadelphia and Washington territories. After graduation from the U. S. Naval Academy in 1919, Mr. Walker subsequently joined D. H. Skeen & Company of Chicago as sales engineer, later becoming vice president. Shortly thereafter he also became president of the Mercon Regulator Company, a subsidiary of D. H. Skeen & Company and in January 1924 resigned to become president of the Illinois Barge Line Company which was later merged with the John I. Hay Company of which he was made vice president and chief engineer. He joined Combustion Engineering Company in December 1934.

Prevention of Clogging with Plate Type Air Heaters

The recent Prime Movers Report on "Boilers, Economizers, Air Heaters and Piping" contains a statement by the Milwaukee Electric Railway and Light Company describing an ingenious method of preventing clogging in its plate type air heaters. Continuous operation without cleaning of four 77,500 sq ft plate type air heaters, when burning midwestern coal of high sulphur content, was made possible by attention to minimum metal temperatures. Referring to the accompanying sketch, the left-hand section illustrates the usual clogging process, while that at the right shows how clogging was prevented by insulation applied at the top to increase the metal temperature. Tests and actual experience have demonstrated that elimination of the low temperature at the inlet end of the plates, by the application of insulation, has prevented initial clogging which, if permitted to start, will extend across the bottom of the heater. The critical metal temperature in economizers and air heaters below which hygroscopic sulphur deposits occur appears to be 210 F.



To Inspect Power Plant of the "Normandie"

Arrangements have been made for a meeting of the Metropolitan Section (New York) of the A.S.M.E. on October 22nd at 6:30 p.m. in the theater of the super-liner "Normandie" which will be in port at that time. Jean Hazard, Chief Engineer of the ship, will present a paper on its propulsion following which there will be an inspection of the power plant. The theater has a capacity of only 400 and passes to the vessel must be obtained from the A.S.M.E. in advance.

Strength and Flexibility of Corrugated and Creased-Bend Piping

By ROBERT L. DENNISON
Lieutenant, U. S. Navy

These investigations were conducted at the U. S. Naval Engineering Experiment Station, Annapolis, Md., under the direction of the Bureau of Engineering and a dissertation on the subject was submitted to Johns Hopkins University in connection with advanced work by the author. This article, which outlines the procedure and summarizes the results, was prepared for COMBUSTION by request. The complete report, including details and mathematical analyses, was submitted to the American Society of Naval Engineers and covers ninety pages in the August 1935 issue of its Journal. The conclusions derived from the investigations are that, while corrugated and creased pipe may be bent to smaller radii than smooth-walled pipe and the forces exerted against anchor points by expansion are low, the factors which operate to increase flexibility also operate to increase the stress, especially where the pipe is subjected to cyclic variable stress.

THE trend toward rising steam temperatures in modern power plants has brought with it many problems of design. One of these concerns suitable means for caring for steam pipe-line expansion. Failure to provide properly for pipe-line expansion may lead to excessively high reactive forces on turbine casings, boiler drums or other anchorage points, leaky flanges or dangerously high stresses in the pipes themselves.

There are several generally accepted methods for providing for pipe-line expansion such as swivel joints, slip joints, expansion joints and configuration of the piping system itself. Swivel joints find application principally with low pressures and temperatures and with small pipe sizes. Slip joints involve the use of packing and stuffing boxes and are seldom used at temperatures exceeding 400 F. Expansion joints are generally incapable of absorbing much movement and are usually installed when rubber or copper might successfully be employed in their construction. For designs involving high temperatures and pressures, expansion is usually cared for by the deformation of the pipe line itself. For this purpose use is made of the familar expansion bend or by changes in direction of the run of piping. If there is considerable expansion to be absorbed and if the reactive forces are to be kept within conservative limits in order not to exceed safe stresses in the piping or to prevent excessive thrust against machinery, the providing of a sufficient number of expansion bends or changes in the direction of piping may require a prohibitive amount of space.

Corrugated piping and creased-bend piping have been proposed as an answer to this problem. Both of these pipe shapes may be bent to a considerably smaller radius than smooth-walled pipe of corresponding diameter. The corrugated shape, and to a lesser extent the creased-bend shape, are more flexible than similar bends formed from smooth-walled pipe. Hence it is possible, using these shapes, to reduce the reactive forces associated with pipe-line expansion. Although several investigators have studied and reported upon the flexibility of corrugated and creased-bend piping there has been a lack of information concerning the strength of such pipe construction.

It may generally be assumed in designing steam pipe lines that the anchorage points are fixed and that all expansion which would result from the heating of the pipe line is forced back into the piping system. The amount of force required to thus deform the piping system is a function of the flexibility of the system.

It has been claimed for corrugated and creased-bend pipe that since shapes bent from piping of this construction are more flexible than corresponding shapes for smooth-walled pipe they consequently may safely absorb more expansion. Many investigators have shown that the corrugated and curved pipe shapes are in fact more flexible than corresponding shapes for smooth-walled pipe. It has been shown by Dr. T. H. von Karman and others that smooth-walled pipe, in the form of an arc, when deflected shows increased flexibility when compared to results of calculations by simple curved bar formulas. This enhancement of flexibility has been proved by Dr. T. H. von Karman and Prof. W. A. Hovgaard to be caused by the flattening of the pipe cross-section to a quasi-elliptical shape when the curvature of the bend is increased. This effect is inappreciable in smooth-walled pipe tangents.

Thus, the use of these flexible pipe shapes appears very attractive. Since, because of their increased flexibility less effort is required to force expansion back into the pipe line, stresses have been assumed to be low, or bends used which have been formed to a smaller radius than would have been possible with smooth-walled pipe.

There are many ways in which a steam line may be stressed in service. Variations in superheat cause con-

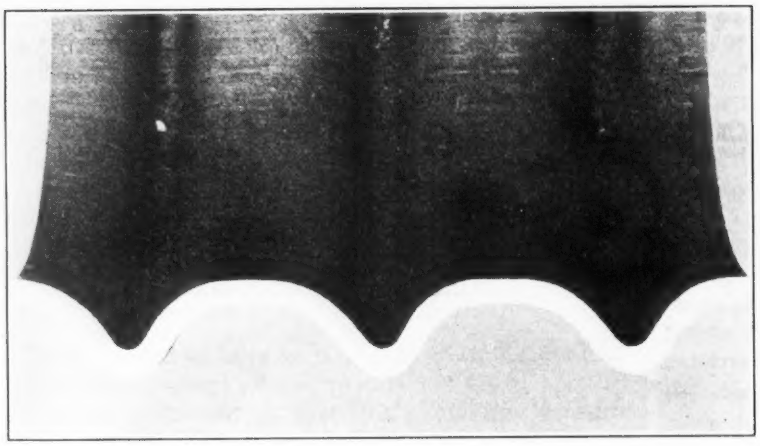


Fig. 1—Longitudinal section showing a profile of typical corrugations in series A piping
Note the slightly different curvatures of the corrugations at the inner pipe surface.

sequent variation in pipe-line expansion with corresponding variation in stress. Changes in pressure likewise create stress cycles. Mechanical vibration of the pipe will induce cyclic stresses as will also relative movement of the anchorage points. The vibration of the piping system may be of small magnitude but when the attendant stresses are superimposed on an existing steady state of high stress early failure may result.

Past practice has based limiting design stresses on purely static stress considerations. If this were a valid basis, a pipe line so designed which would withstand the initial application of maximum temperature and maximum pressure would thereafter, barring corrosion, be incapable of failure. A static failure of ductile material is usually preceded by distortion, buckling and collapse. Failures in steam pipes, however, do not generally occur in this manner. Here cracks develop leading ultimately to rupture. Quite obviously a usual piping system is not subjected to the type of loading, such as a bridge member receives, which will lead to buckling and collapse. It is, however, usually subjected to cyclic variations of stress leading to fatigue fracture. For these reasons the fatigue limit rather than the yield point or ultimate strength would appear to be a more logical basis for design. The fatigue limit of the metal used in pipes may be readily determined.

The question of when this limiting stress is exceeded in the pipe itself is exceedingly complex. The manner of the manufacture of the corrugated and creased pipe shape results in each corrugation or crease being an individual, each closely alike but not identical with its neighbor. The rigid mathematical analysis of the stresses existing in these shapes appears incapable of accomplishment.

Scheme of Investigation

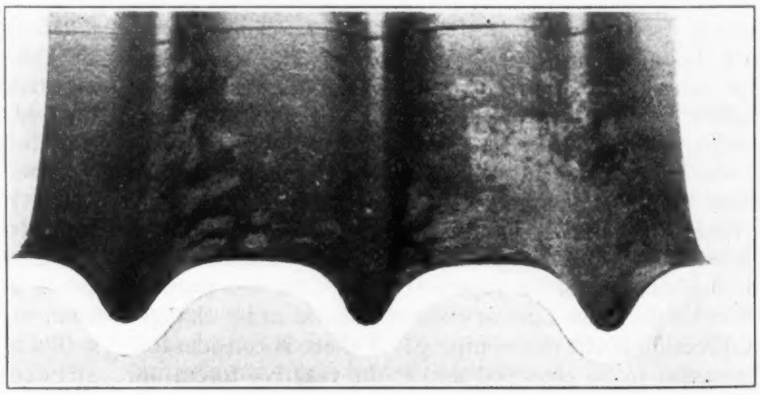
The presence of corrugations or creases in a pipe while increasing the flexibility also acts to increase the stress. If fatigue tests are made on these pipes so as to determine an endurance limit for each given type, it will develop that the stress at this endurance limit when computed from the ordinary Rankine formulas of mechanics is far different from the stress corresponding to the known endurance limit of the steel. For the purpose of determining the stress at which the pipes were operated in fatigue tests the bending formula was used

$$S = \frac{Mc}{I}$$

In this formula, M is the value of the applied bending moment and I/c is the value of the section modulus of the basic pipe. In order to determine this value the moment of inertia, I, of the cross-section was computed from dimensions of the pipe before being corrugated or creased. The value of c was likewise taken as the radius of the outer surface of the basic smooth-walled pipe. This equation then yields values for stress, S, which are nominal values. The term "nominal stress" has been applied to this quantity. The ratio of the endurance limit for the steel to the endurance limit of the pipe, when the latter is expressed in terms of nominal stress, will then be a direct indication of the stress multiplication effect due to the corrugated and creased-bend shapes. The writer has called this factor the "stress intensification factor" and has given it the designation N.

In addition to the determination of the stress intensification factor, a method was developed for determining the stress distribution in pipes from internal pressure

Fig. 2—Longitudinal section showing a profile of typical corrugations in series B piping
Note here also the lack of exact similarity in the corrugation.



and from applied bending moments. This method employs the use of strain gages applied to the outside of the pipe wall and involves the calculation of the critical stresses at the inner pipe wall surface derived from these external strain measurements. A further description of this method will not be included here.

Material under Investigation

This investigation covers three classes of pipe shapes, all constructed from 6-in. seamless steel tubing as follows:

Series A—Corrugated piping with nominal wall thickness 0.280 in. and shape of corrugation as shown in Fig. 1.

Series B—Corrugated piping with nominal wall thickness 0.288 in. and shape of corrugation as shown in Fig. 2.

Series C—Creased-bend piping of a typical shape of crease as shown in Fig. 3 and with nominal wall thickness 0.288 in.

Complete tabulations of the pipe shapes investigated under each of the above classifications, together with details of heat treatments and physical properties, are given in Figs. 4 and 5.

In manufacturing a corrugated pipe the seamless steel tubing is placed in a press between a fixed and movable head. A gas-fired ring furnace is placed around the pipe and a narrow band heated. When the pipe has reached a proper temperature, it is compressed along its axis by the press, thus raising a corrugation in the heated section. When the compression has raised the corrugation a desired amount, the press is stopped, the corrugation quenched and the furnace moved on to form the adjoining corrugation. The first motion of the press in forming a corrugation causes an upsetting of the metal in the heated band before swelling starts. This action accounts for the increased wall thickness to be found in the radial section through the crest of the corrugation. If a bend is to be formed, the bending is done after the corrugating process. The fabrication is followed by a suitable annealing heat-treatment.

In forming a creased bend, the pipe is first filled with sand, locally heated at each crease in turn and bent about 10 deg at each crease. The creases thus bulge out from the original pipe wall and disappear at about two-thirds of the circumference of the pipe. This method is said to have the advantage of permitting bending without a thinning of the wall on the outside of the bend and a thickening of the wall on the inside of the bend. Both corrugated and creased-pipe bends may be made with much smaller radii than may be formed from corresponding smooth-walled tubing. The processes of manufacture are patented.

It is to be carefully noted that the method of manufacture results in each corrugation or crease being an individual with possibly different characteristics than its neighbors. Non-concentricity of the bore of the tubing with the outer surface may be serious if it provides a minimum section in the plane of maximum stress. Because of the localized stress conditions in these pipes, it is also highly important that no local defects such as inclusions, laps, seams or surface cracks appear in the tubing because of the danger that one of these areas of weakness occur in a location of maximum stress.

Testing Machines

Two types of testing machines were used to determine the endurance properties of the pipes. One type of machine was designed to apply a specific stress to the specimen and the other type a specific strain. The specific stress machine was, in principle, a large scale model of the rotating beam fatigue machine frequently used in the laboratory for determining the endurance limit of small specimens. The specific strain machines were designed to subject the pipe to alternating flexure by applying a constant deflection of the specimen. A total of four machines of both types was built.

A view of the specific stress rotating beam testing machine is given in Fig. 6. Naturally only the long tangent specimens could be adapted to this apparatus. As will be seen from the figure, the long tangents were mounted in trunnion bearings at either end. Babbitt metal journals cast on the pipe carried collars which supported the desired dead weight load. The pipes were rotated by a motor and belt drive through a universal joint. With equal loads applied to the arms hanging from each collar, the bending moment across the corrugated section of the tangent was very nearly uniform. Hence, all corrugations between the journals were subjected to approximately equal stress, and failure could reasonably be expected to occur in the weakest corrugation. With the pipe section at rest under load, the stress distribution in the pipe wall may be considered to be symmetrical with respect to a horizontal plane through the longitudinal neutral axis of the pipe. Now as the pipe is rotated 180 deg, the fibers which were in the top half-section under certain states of stress, will acquire an equal value of stress but with opposite sign. This same reasoning applies to the fibers originally in the bottom half-section. Thus, for a complete revolution of the pipe each fiber

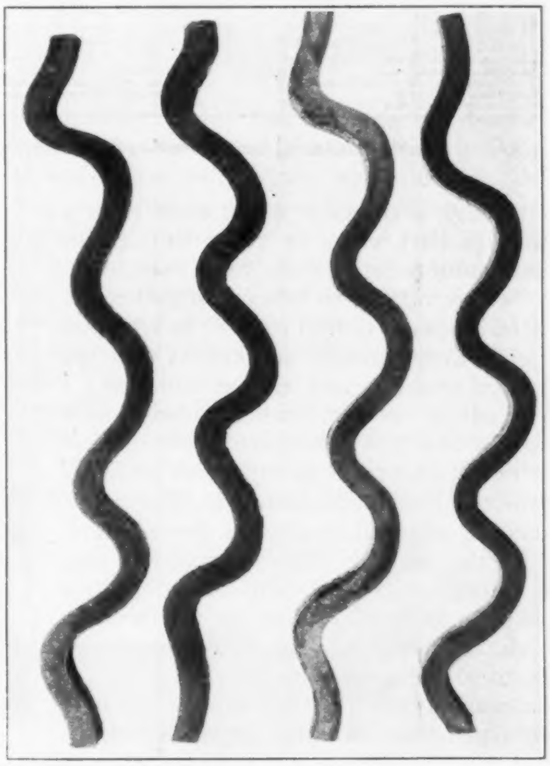


Fig. 3—Longitudinal strips cut from the plane of bending in a creased-bend

These strips were cut from an expansion U-bend. Each strip represents half an arc length of each of the four creased arcs. In this photograph the inner pipe surface is at the right of each strip. Note the individual character of the creases.

Type Bend	Designation	R	A	B	C	D	Class Steel A.S.T.M. Spec A-106-33T	Tensile Strength	Yield Point	Elongation % 1 in	Elongation % 2 in	Red Area %	C	S	P	Mn	Si	Heat Treatment (See Note)
Straight Tangent	AA		6'-5⅜"	5'-9⅝"			A	51,500	35,500	62	43	67						Normalized
	AB		6'-4⅜"	5'-8⅜"			A	51,650	34,350	62	42	68.4	0.13	0.024	0.011	0.53	0.01	Stress Relieved
	AC		6'-5"	5'-8			B	68,200	43,900	51	36	43.9	0.27	0.029	0.014	0.56	0.073	Normalized
Expansion U-Bend	AD	1'-9"	8'-4⅜"	4'-1⅜"	4'-0¼"		A	58,000	35,350	64	43.5	63.1						Normalized
	AE	1'-9"	8'-6⅜"	4'-0¼"	4'-0¼"		B	70,000	43,000		36	46						Normalized
	AF	1'-9"	8'-6⅜"	4'-0⅜"	3'-11⅜"		A	56,700	38,100	64	43.5	65.9	0.16	0.022	0.015	0.54	0.10	Stress Relieved
Quarter Bend with Tangents	AG	1'-9"	5'-3⅜"	5'-3⅜"			A	52,300	33,400	51	33	62.9	0.13	0.024	0.011	0.53	0.01	Normalized
	AH	1'-9"	5'-3⅜"	5'-3⅜"			A	51,600	37,200	68	48	72.6						Stress Relieved
Straight Tangent (Plain Ends)	AJ		13'-10"	3'-4"			A	60,000	36,900		41.7							Stress Relieved
	AK		13'-9⅜"	3'-4"			A	61,250	40,000		39							Stress Relieved
	AL		13'-10"	3'-10"			A	61,500	42,000	60	41.5	61						Stress Relieved
Single Offset Quarter Bend	AM	1'-4"	3'-11½"	2'-4"	1'-0¼"	1'-6¼"												
		Special Semi-Corrugated Bend 6.125 in OD × 0.173 in Wall Thickness																

All Pipe - Seamless Steel Tubing 6.625 in O.D. × .280 in Wall Thickness
Pitch of Corrugations - 2½ in
Amplitude of Corrugations - ½ in

Note - Heat Treatment
Normalized - 1575°F for 30 min. Cooled in Air
Stress Relieved - 1200°F for 30 min. Cooled in Furnace

Fig. 4—Material list for series A piping

mitted to one end of the cylinder and discharged from the other through an arrangement of plug cocks whose action was governed by a linkage operated from an electric motor. With the apparatus in operation, the pump rod would move back and forth at a cycle frequency of approximately 20 per minute. The amount of rod movement could be adjusted by suitable setting of the linkage mechanism to the plug cocks and could be positively limited by adjustable stops attached to one of the pump tie rods. The outer end of the pump rod was attached to a pipe flange for endurance testing purposes. This machine was used for testing certain expansion U-bends only.

Endurance Limit for the Steel

It was not considered practicable to machine satisfactory fatigue-test specimens from the sample pipes used in the investigation. It is not necessary, however, to determine the actual

passes through a complete stress cycle from a maximum positive value of stress through an equal minimum negative stress value back to the positive stress value. In this investigation the pipe rotative speeds ranged from 80 to 220 rpm. The speeds were carefully selected so that the pipe rotated smoothly and without vibration.

Two kinds of specific strain machines were developed. One type consisted of a driven shaft that carried at either end an adjustable eccentric upon which a journal was mounted. The eccentric could be adjusted to give the desired crank throw for the journal. This journal was connected through a vertical connecting rod to a suitable point to apply load on the pipe section. The cycle frequency in these machines ranged from 30 to 90 cycles per minute.

The remaining specific strain type of machine was developed from the water end of a reciprocating pump. The pump was mounted with its rod horizontal and with a hydraulic system, under pressure, connected to either end of the pump cylinder. The water was alternately ad-

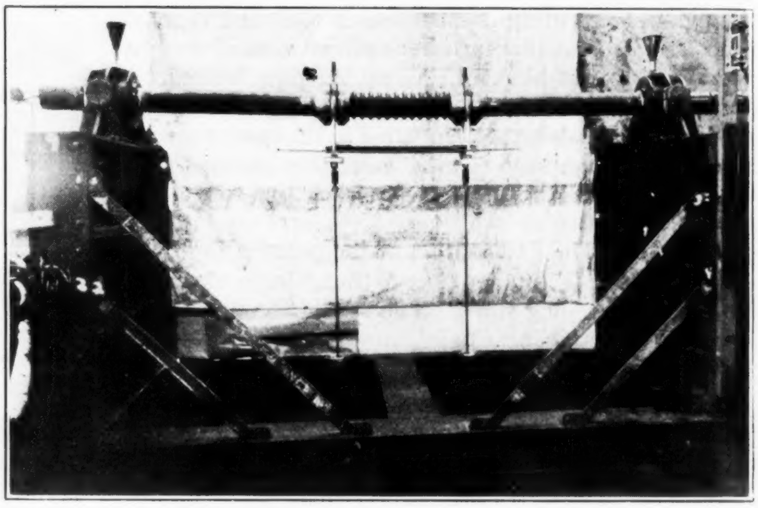


Fig. 6—The rotating beam testing machine

This machine was designed to apply a constant loading, or specific stress, to the pipe specimen. The pipe was rotated about its axis by a belt drive. A desired amount of dead weight load could be applied on the hangers shown below each journal at the ends of the corrugated section. Note the six-foot scale extended below the corrugated pipe section.

Type Bend	Designation	R	A	B	C	D	Class Steel A.S.T.M. Spec A-106-33T	Tensile Strength	Yield Point	Elongation % 1 in	Elongation % 2 in	Red Area %	C	S	P	Mn	Si	Heat Treatment (See Note)
Straight Tangent	BA		6'-3⅜"	4'-9⅜"			A	53,800	37,200	48.0	35.8	33.5						Stress Relieved
	BB		6'-3⅜"	4'-11⅜"			A	53,800	37,200	48.0	35.8	33.5	0.14	0.029	0.012	0.39	0.01	Stress Relieved
	BC		6'-4"	4'-10⅜"			A	53,800	37,200	48.0	35.8	33.5						Stress Relieved
Quarter Bend with Tangents	BD	1'-3"	5'-3"	5'-3"			A	55,300	37,400	55.0	42.0	54.8						Stress Relieved
	BE	1'-3"	5'-3"	5'-2⅜"			A	53,750	38,300	58.0	36.5	67.4						Stress Relieved
	BF	1'-3"	5'-3"	5'-3"			A	53,750	38,300	58.0	36.5	67.4						Stress Relieved
Straight Tangent (Plain Ends)	BG		13'-10"	3'-4"			A	61,100	42,100	61.0	41.0	63.4						Tempered
	BH		13'-10"	3'-4"			A	72,600	46,000		25.5	75.1	0.17	0.024	0.015	0.64	0.15	Tempered
	BJ		13'-10"	3'-4"			A	57,500	38,300	59.0	37.5	59.8						Stress Relieved
Quarter Bend with Tangents	CA	1'-3"	5'-3"	5'-3"			A	59,900	41,900		41.0	66.1	0.16	0.022	0.015	0.64	0.15	Stress Relieved
	CB	1'-3"	5'-3"	5'-3"			A	53,800	35,250		40.2	60.0						Stress Relieved
	CC	1'-3"	5'-3"	5'-3"			A	56,500	40,000	60.0	39.0	61.7						Stress Relieved
Expansion U-Bend	CD	1'-3"	8'-3⅜"	3'-11⅜"	4'-0⅜"	0'-11⅜"	A	63,000	48,100		32.8	68.6						Stress Relieved
	CE	1'-3"	8'-6⅜"	4'-0⅜"	4'-0⅜"	1'-0"	A	60,000	43,700	59.0	42.0	66.9						Stress Relieved
	CF	1'-3"	8'-6⅜"	4'-0"	4'-0"	1'-0⅜"	A	60,500	42,300	46.0	30.5	60.6						Stress Relieved

All Pipe - Seamless Steel Tubing
6.625 in O.D. × 0.288 in Wall Thickness
Pitch of Corrugations - 2½ in.
Amplitude of Corrugations - ½ in.
Pitch of Creases - 11° 15' approx.

Note - Heat Treatment
Stress Relieved - Heated to 1200°F for 18 min. Cooled in Air.
Tempered - Heated to 1800°F. Quenched in Water
Reheated to 1200°F. Cooled in Air.

Fig. 5—Material list for series B and C piping

endurance limit of this steel by direct test. It is generally accepted that a metal's endurance limit under cycles of reversed flexural stress may be expressed as a definite percentage of its ultimate tensile strength. No such correlation has been discovered between the endurance limit and any other physical property of a metal. The ratio of the endurance limit to the ultimate tensile strength has been termed the endurance ratio by Dr. D. J. McAdam, Jr., and for most steels will have a value ranging between 0.40 and 0.50. Fig. 7 gives a fatigue graph for a carbon steel whose composition was similar to the steel used in the pipe. This steel has a tensile strength of 63,000 lb per sq in. and an endurance limit of 27,000 lb per sq in. The value of the endurance ratio is 0.43 which has been applied to the observed tensile strength of the pipe steel in order to determine the value of its endurance limit.

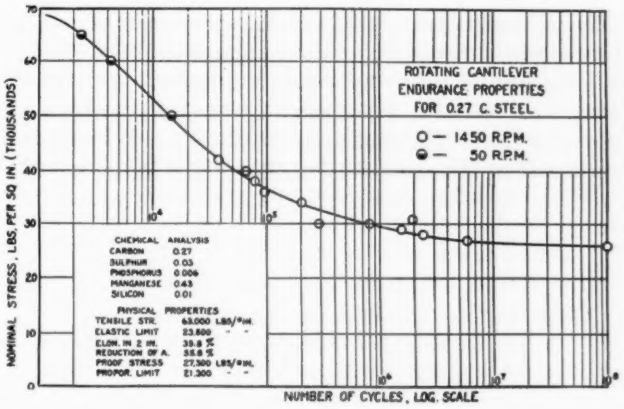


Fig. 7—Fatigue graph for a low-carbon steel similar to that used in the pipe under test

The ordinates represent the applied bending stress and the abscissa the number of cycles. The points on the graph represent failure of specimens.

In connection with Fig. 7 it may be stated that the three points of highest stress on the fatigue graph were determined at a specimen speed of 50 rpm, while the points for the remainder of the graph were determined at 1450 rpm. The point representing the highest stress on this graph was obtained with a computed bending stress on the specimen actually greater than the ultimate tensile strength.

Load-Deflection Characteristics and End Conditions

Since the specific strain type of fatigue testing machine imposes a definite deflection on the pipe, the bending moments and the consequent applied nominal stresses could be determined only from a knowledge of the relationship between the load and deflection of the specimen under consideration. In order to establish the load-deflection characteristics, each pipe shape was mounted with one end firmly secured to a heavy steel foundation plate and dead weight loads applied to the other end. Deflections were measured by means of micrometers with increments of applied load. Small balls, acting as gage points for the measurements, were attached to the pipe flanges and to battens which were fixed to the foundation. Fig. 8 gives a view of a typical arrangement for the load-deflection determination.

Consider a pipe bend with its end flanges firmly fixed and subjected to an increase of temperature. The expansion of the pipe wall will cause the flanges to exert certain forces and moments on the anchorages which may be resolved into a force perpendicular to the flange, a force parallel to the flange in the plane of the bend and a couple also in the plane of the bend. This is called a fixed-end condition. If the two forces continue to act but the couple is removed, the flange is thus permitted to rotate and the condition is known as free end.

The fixed-end condition was simulated in load-deflection testing as shown in Fig. 8. Here the restraining force parallel to the flange in the plane of the bend is clearly zero. It is known from mechanics that a force and a couple acting in the same plane may be replaced by a single force whose moment arm is equal to the value of the couple divided by this force. It will be seen from the photograph that this is accomplished by the method of loading shown. Here, as the load is applied, the flange moves in parallel planes. This condition was checked on test by a spirit level mounted on the upper flange. The proper value for the moment arm was first computed. Any subsequent minor adjustment necessary was indicated by the level.

There is a slight discrepancy between the state of affairs governing a load deflection or fatigue test and the conditions actually obtaining in service. In the test conditions the forces and couples are applied on the cold or unexpanded pipe, hence on a pipe shape slightly different, from a dimensional standpoint, from the same pipe under elevated temperatures in service. This difference, however, is so slight that it may be disregarded for all practical purposes. It may be remarked that pipes in service will probably operate at one condition intermediate between the free-end and fixed-end conditions, and that the nature of these end conditions will have a decided effect on the stresses produced in the pipe itself.

Results

The flexibility factor for all tangents tested was slightly greater than six. As has been indicated, a smooth-walled arc will flatten when loaded and the flexibility will thus be increased. For this reason the corrugated arc does not show as great a relative flexibility to an actual smooth-walled pipe arc as is the case of the tangents. For the arcs, this corrugated arc appears to have an advantage over the smooth arc of about 2.6 to 1. The creased arc appears to have a relative flexibility of about 1.3 to 1.

The results of the fatigue tests of each of the series of corrugated and creased piping are shown graphically in Figs. 9, 10 and 11. The scheme used in plotting the fatigue graphs is the usual one for presenting results of

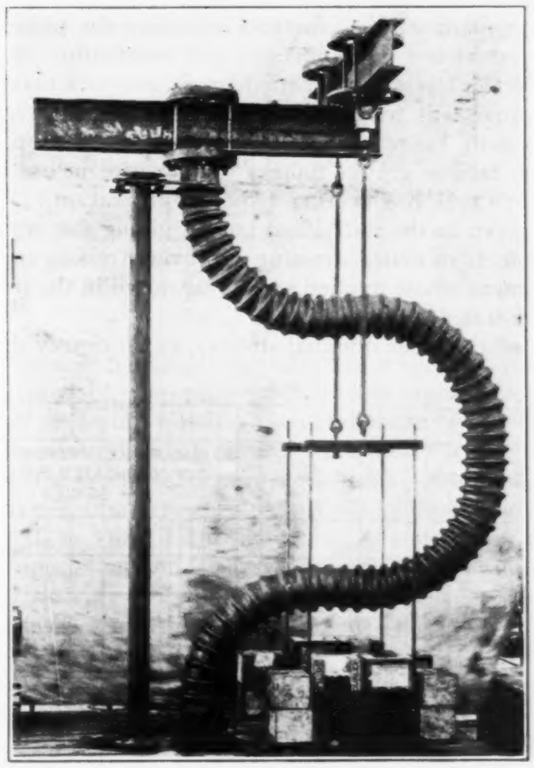


Fig. 8—An expansion U-bend erected for load-deflection measurements in the fixed-end condition

The moment arm from the center of the top flange to the line of action of the applied dead-weight load was adjusted so that the top flange always remained horizontal as increasing loads were applied. This condition was checked by a spirit level (not shown) mounted on the top flange. The deflections were measured from the top flange to a fixed framework just under the flange.

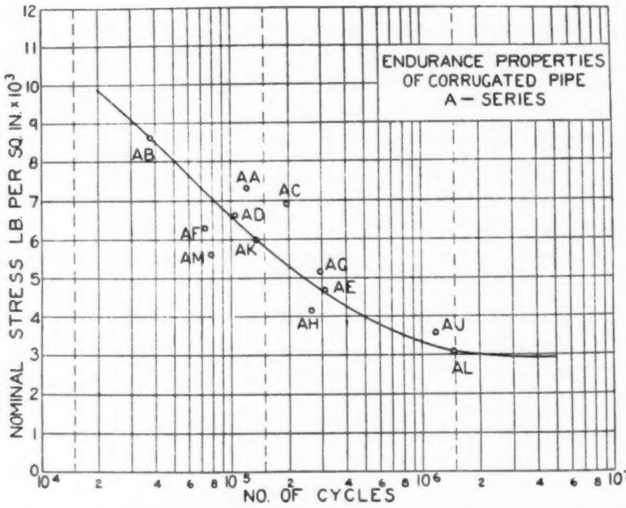


Fig. 9—Endurance properties of corrugated pipe, series A

fatigue testing. The axis of ordinates is scaled uniformly to represent the semi-range of alternating stress while the axis of abscissas is scaled logarithmically to represent the number of cycles. The points on the graph, in general, indicate failures of the designated pipe shapes. In some cases the applied nominal stress values were raised during the test. Where this was done paths are shown on the graph to show the manner in which the specimen was tested at increasing stress values. Where arrows alone are attached to points this does not represent failure but means that the specimen is to be considered as continuing to run beyond the plotted point. The bending moments and the nominal stresses were computed at the point of actual fracture for each specimen. The graphoanalytical method involving the principle of superposition was employed in this calculation. In cases where the stress cycle varied from zero to a maximum, the range was transformed for plotting purposes, to an equivalent range for alternating stress. The plotting of the fatigue graphs follows the practice in use at the U. S. Naval Engineering Experiment Station. Weight was given to the individual points, giving due regard to the effects of under-stressing and over-stressing on those specimens whose applied stress was raised in the progress of the test.

In all cases the nominal stresses, as previously defined,

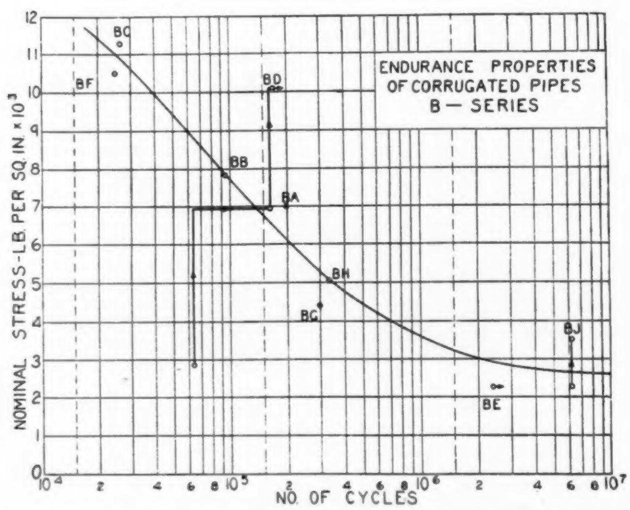


Fig. 10—Endurance properties of corrugated pipe, series B

were computed from the simple beam formula as the stresses which would be produced in a theoretical smooth-walled pipe of the same basic dimensions as the corrugated or creased pipe when subjected to the same applied forces. The longitudinal stresses only were computed and in the comparative theoretical smooth-walled bend no allowance was made for any increase of stress caused by flattening. The direct stresses resulting from the loading are a small percentage of the stress due to bending and have, therefore, been neglected as minor in computing the nominal stresses.

Load-deflection tests made on pieces *AD*, *BE* and *CF* without internal pressure and with 400 lb per sq in. internal pressure showed practically no difference. The flexibility of these pipes was unaffected by the internal pressure.

These three pipe shapes were each fatigue tested with 400 lb per sq in. internal pressure. Four failures resulted from these tests. Piece *AD*, a corrugated U-bend, provided one failure. Piece *BE*, a quarter-bend with creased arc and corrugated tangent leg, failed in the creased arc, was repaired by welding and later failed in the corrugated leg. Piece *CF*, a creased U-bend, provided the fourth

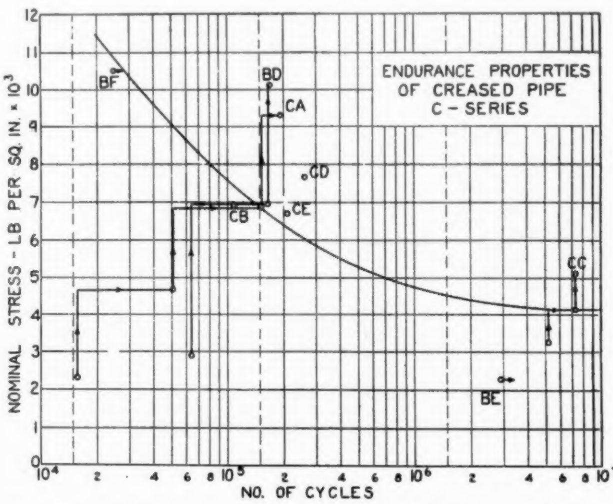


Fig. 11—Endurance properties of creased pipe, series C

failure. In each case, disregarding the stresses from internal pressure and referring the failures to the proper fatigue graph for the applied nominal stresses induced by the bending moments, the points of failure lie in close proximity to the curves. These results indicate that the internal pressure had no appreciable effect on the endurance properties of the bends thus tested.

Stress Intensification Factor

The endurance limit is a physical property of a metal and has a definite value. In the case of the steel used in the manufacture of the piping under investigation, the endurance limits are as follows:

Series A—25,100 lb per sq in.
Series B—24,600 lb per sq in.
Series C—25,300 lb per sq in.

These values were determined as 43 per cent of the average tensile strength of the specimens in each series of pipes.

In the asymptotic portion of the endurance graphs, the stress in the metal must be at the endurance limit. The stress intensification factor has been defined as the ratio between the endurance limit and the nominal stress at this point on the fatigue graph. The values of the stress intensification factors obtained in this manner are:

Series A, *N*—8.7
Series B, *N*—9.5
Series C, *N*—6.1

These values indicate that the nominal stresses have been effectively multiplied at the endurance limit by the factors indicated. It is considered that the discrepancy between the Series A and Series B pipes may partially be explained by the relatively few points available to determine the endurance limits on the fatigue graphs.

Safe Working Stresses

In the great majority of practical applications a piping system will not be called upon to undergo the number of cycles of stress equivalent to that at which the endurance limit is determined. In the lifetime of an ordinary power plant, excluding serious vibration, the number of stress cycles imposed on the piping will probably be numbered in thousands rather than millions. It seems unreasonable then, in most cases, to use in design a stress value corresponding to the true endurance limit.

The true endurance limit represents the limiting value of stress to which a metal may be subjected in order to withstand an indefinitely large number of stress cycles. The fatigue graph defines this limit and also indicates the number of stress cycles to cause failure at stresses greater than the endurance limit. The value of stress taken from a fatigue graph for a stated number of cycles is here defined as the virtual endurance limit for the number of cycles selected. The definition for the stress intensification factor is extended to include the ratio of the virtual endurance limit of the steel to the virtual endurance limit of the pipe, the latter value again being expressed in terms of nominal stress.

The method used in determining the virtual endurance limits of the steel was similar to that described for determining the true endurance limits. At various selected cycle values, the stresses read from the fatigue graph of Fig. 7 were expressed as certain percentages of the tensile strength of the steel used in determining that fatigue curve. These percentages were then applied to the experimentally determined tensile strengths of the pipe steel to establish the virtual endurance limits at the selected cycle values. This procedure was followed considering each series of piping individually. The stress intensification factors were then readily determined by dividing the virtual endurance limits of the steel by the virtual endurance limits of the pipe at selected cycle values. In calculating the safe working stresses a method involving recognition of both the yield point and the endurance limit was used. (For a complete description of this method see paper by C. R. Soderberg, "Factor of Safety and Working Stress," APM-52-2 Transactions of Amer. Soc. Mech. Eng., 1930; also, "Strength of Materials," Vol. II, by Timoshenko.) The safe working stress was computed for two conditions: one assuming that the factor of safety was 2, and the other that the factor of safety was 1.5. The results of these computations are tabulated below:

SAFE WORKING STRESSES FOR SERIES A—CORRUGATED PIPE

Average Tensile Strength = 58,400 lb per sq in.
Average Yield Point = 38,200 lb per sq in.

No. of Cycles	Stress Intensification Factor N	Virtual End. Limit σ_n	Safe Working Stress for $f = 2$ $\sigma_{max.}$	for $f = 1.5$ $\sigma_{max.}$
1×10^7	8.7	25,100	2680	3,580
5×10^5	7.2	28,800	3640	4,840
1×10^5	5.2	34,000	5580	7,430
5×10^4	4.7	37,500	6590	8,800
3×10^4	4.5	40,600	7300	9,720
2×10^4	4.4	43,500	7860	10,460

SAFE WORKING STRESSES FOR SERIES B—CORRUGATED PIPE

Average Tensile Strength = 57,300 lb per sq in.
Average Yield Point = 39,100 lb per sq in.

No. of Cycles	Stress Intensification Factor N	Virtual Endurance Limit σ_n	Safe Working Stress for $f = 2$ $\sigma_{max.}$	for $f = 1.5$ $\sigma_{max.}$
1×10^7	9.5	24,600	2430	3,240
5×10^5	6.4	28,300	3980	5,300
1×10^5	4.3	33,400	6480	8,620
5×10^4	3.9	36,800	7620	10,160
3×10^4	3.8	40,000	8280	11,040
2×10^4	3.7	42,600	8880	11,840

SAFE WORKING STRESSES FOR SERIES C—CREASED PIPE

Average Tensile Strength = 59,000 lb per sq in.
Average Yield Point = 40,900 lb per sq in.

No. of Cycles	Stress Intensification Factor N	Virtual Endurance Limit σ_n	Safe Working Stress for $f = 2$ $\sigma_{max.}$	$f = 1.5$ $\sigma_{max.}$
1×10^7	6.1	25,300	3770	5,020
5×10^5	5.5	29,000	4670	6,220
1×10^5	4.5	34,300	6430	8,570
5×10^4	4.2	37,800	7370	9,820
3×10^4	3.9	41,000	8340	11,100
2×10^4	3.8	43,800	9010	12,000

These results indicate that, in general the Series B corrugated pipe could be subjected to slightly higher safe working stresses than the Series A pipe. All pipes were manufactured with the same amplitude and pitch of corrugations. The Series B pipe, however, had a slightly greater wall thickness and a somewhat different curvature of corrugations from the Series A pipe. It should be noted here that the flexibility factors for the tangent lengths of piping of each of the two series are essentially the same.

RATIO OF TENSILE STRENGTH TO SAFE WORKING STRESS

Cycles	Series A Corrugated $f = 2$	$f = 1.5$	Series B Corrugated $f = 2$	$f = 1.5$	Series C Creased $f = 2$	$f = 1.5$
1×10^7	21.8	16.3	23.5	17.7	15.7	11.7
5×10^5	16.0	12.1	14.4	10.7	12.6	9.5
1×10^5	10.5	7.9	8.8	6.6	9.2	6.9
5×10^4	8.9	6.6	7.5	5.6	8.0	6 0
3×10^4	8.0	6.0	6.9	5.2	7.1	5.3
2×10^4	7.5	5.5	6.5	4.8	6.5	4.9

The average tensile strength for the steel used in the manufacture of the pipes tested was approximately 58,000 lb per sq in., which is about the mean of the range of tensile strengths of steels used in ordinary commercial practice for seamless drawn steel tubing intended for high temperature service. (See A.S.T.M Specifications A-106-33T.) These ratios may then be used as a satisfactory approximation for design purposes in determining the safe working stress for pipes fabricated from tubing whose tensile strength and yield-point values are within this usual commercial range.

The use of a factor of safety of 2 is considered conservative. The use of a lower value, such as 1.5 or some intermediate value, would undoubtedly be justified when it was known that the tubing material was of uniformly high quality, when the service conditions were not severe, and the values of the applied stresses could be predicated with reasonable accuracy.

Pipe shapes may successfully withstand large deflections and absorb very high applied stresses for single, or

very few, applications of load. There may be some piping installations where the service application may involve an approximately steady loading condition. When this is true the piping design may be based entirely on the static tensile properties of the metal. If, however, in the designer's opinion, the piping is to be subjected to variable stresses, static strength considerations are insufficient and the design should be based on an extremely liberal allowance for the number of stress cycles to be encountered.

The fatigue graph is useful primarily for determining the safe limiting value of stress for a large number of

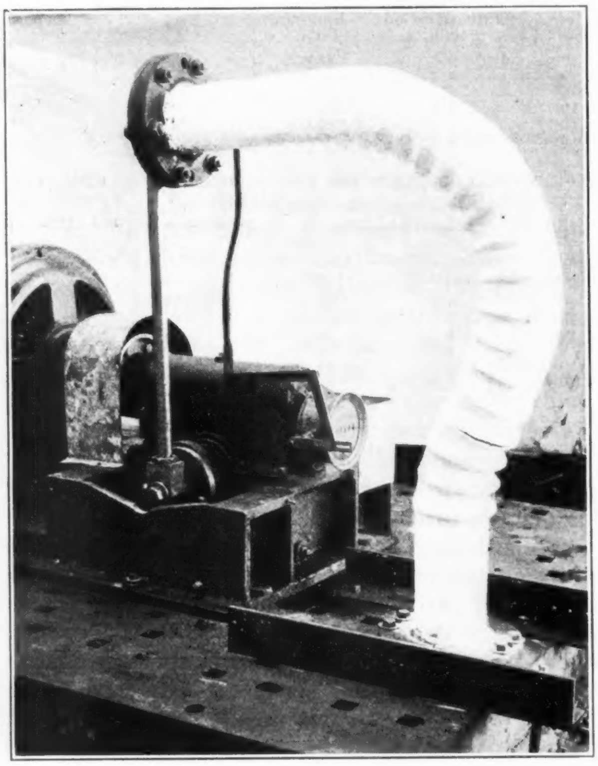


Fig. 12—The single offset quarter bend as erected for test in a specific strain type fatigue testing machine
Note the crack, as a result of the fatigue test, which appears in the fourth corrugation from the bottom of the pipe.

cycles. The curve is necessarily the mean path through a number of observed points and cannot be considered as applying with sufficiently close accuracy for literal application in predicting individual failures. A slight variation in stress value will cause a large percentage error in the number of cycles at any point on the curve. These considerations are particularly important in the high-stress, low-cycle region of the fatigue graph. The action of steel at high variable stresses and high temperatures cannot be safely predicated. For these reasons the writer is of the opinion that 20,000 cycles of variable stress should be the absolute minimum basis for design purposes when cyclic stress conditions are considered, regardless of the actual predicated numerical value of the stress cycles to which the pipe may be expected to be subjected. In any case the number of cycles selected as a design basis, in accordance with the method presented, should be considerably greater than the number of cycles it is estimated the pipe will experience. This is particularly important, as has been previously indicated, in applications where corrosion is likely to be a factor.

General Summary

Although many of the conclusions reached in this investigation are of general application, the values given for flexibility factors for limiting design stresses will not be strictly applicable where piping is being considered which departs from geometrical similarity with the pipes tested. A flexibility factor value of between 5 and 6 will probably closely fit any usual corrugated pipe. The true value, however, can easily be determined by simple experiment.

The use of corrugated and creased pipe offers two principal advantages: the forces exerted against the anchor points by pipe expansion are low, and the pipe may be bent to much smaller radii of curvature than is practicable for smooth-walled pipe. However, the factors which operate to increase the flexibility of these pipes in turn operate to increase the stress. This increase of stress may more than counterbalance the effect of increased flexibility.

Acknowledgments: The experimental work of this investigation was undertaken at the U. S. Naval Engineering Experiment Station, Annapolis, Maryland, under the authorization and direction of the Bureau of Engineering, Navy Department. The writer wishes to acknowledge his indebtedness to Captain O. L. Cox, U. S. Navy Director of the Experiment Station, to Mr. W. C. Stewart, Metallurgist, and to Mr. B. F. Treat, Associate Mechanical Engineer, both of the Experiment Station Staff, and to Dr. W. M. Coates, Associate Professor of Mathematics and mechanics at the U. S. Naval Post-graduate School, to Professor A. G. Christie of Johns Hopkins University and to others of his associates at the Naval Experiment Station who have assisted him.

Field Demonstration of Control Equipment

A new field demonstration service, directed at industries where instrumentation and control are major problems, was recently put into effect by the Mason-Neilan Regulator Company of Boston, Mass. The new service unit, a specially designed truck, completely equipped with up-to-the-minute reducing valves and control instruments for pressure, temperature and flow, is making a nation-wide tour, giving demonstrations of modern control in important chemical plants, paper mills, refineries, process and power plants. New York, Philadelphia, Pittsburgh, Toledo, Chicago, St. Louis, Tulsa and Houston are some of the larger cities included on the itinerary.

With the retirement of F. P. Cox as manager of the West Lynn, Mass., works of the General Electric Company on September 1, Nelson J. Darling, manager of the River Works of the company at Lynn, assumed management of both plants. Mr. Darling in his new duties will have N. M. DuChemin, formerly superintendent of the West Lynn works, as assistant manager in charge of operations at West Lynn.

Sixteen coal companies having mines in Harlan County, Kentucky, have filed suit in the U. S. District Court for western Kentucky attacking the constitutionality of the recently enacted federal coal legislation, commonly known as the Guffey Coal Bill. The tax feature is being stressed in the complaint.

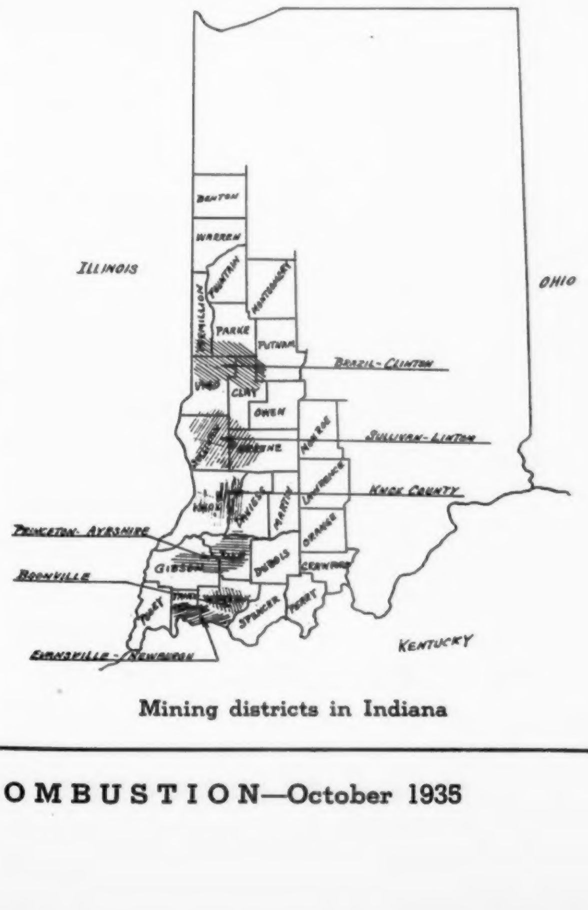
INDIANA COALS—
Their Classification and Analyses

By P. B. PLACE

Combustion Engineering Company, Inc.

This is the sixth article of the series in which the author discusses the bituminous coals by states, identifying those produced in various counties and from different seams by their trade names and giving their characteristics. To date Ohio, Kentucky, Virginia, Illinois and Indiana coals have been covered, as well as a general survey of the coals of the United States which appeared in the first article. In Indiana, three types of coal are found, namely, midwest bituminous, block coal which is a good domestic fuel because of its low sulphur and ash content, and cannel coal.

THE Indiana coal fields lie in some twenty counties in the southwestern part of the state. The area is bounded on the west by central and southern Illinois and on the south by western Kentucky. It is part of the Eastern Interior Coal Region which covers Illinois, Indiana and western Kentucky and is divided into six mining districts. The map shows the portion of the state that is underlain with coal and the location of the districts. The latter are listed in Table I together with the counties in the districts and the principal seams mined.



Mining districts in Indiana

Indiana has twenty or more coal seams but the bulk of the production comes from only three, the Fifth Vein, the Fourth Vein and the Sixth Vein. The total production of the state is only four or five per cent of the bituminous production of the United States and eighty per cent of it comes from six counties, namely, Pike, Greene, Vigo, Knox, Sullivan and Vermillion. Tables II and III give the relative production in the principal states and counties in Indiana during recent years.

TABLE I

COAL MINING DISTRICTS IN INDIANA

Name of Mining District	Counties Included	Principal Seams Mined
Princeton-Ayrshire	Pike Gibson	5th
Sullivan-Linton	Greene Sullivan	5th 4th 6th
Brazil-Clinton	Vigo Vermillion Owen Clay	5th 3rd 4th 6th Minshall Block
Knox	Knox	5th 4th
Boonville	Warrick Spencer	5th
Evansville-Newburg	Vanderburg	5th

All of the coal seams in Indiana are found in the Pennsylvanian Series which is divided into three formations for geological classification. These divisions and the principal seams found in each are listed in Table IV.

Three types of coal are found in Indiana, the common midwest bituminous, a block coal and a cannel coal. The bituminous constitutes the bulk of the state's production and is similar to western Kentucky and Illinois coal in appearance. It has a banded structure of alternate bright and dull coal with frequent partings of shale, clay or pyrite. Thin layers of mineral charcoal are common. The coal mines out in fairly large blocks

TABLE II
RELATIVE PRODUCTION OF BITUMINOUS COAL
IN PRINCIPAL COAL PRODUCING STATES
1929 - 1932
PER CENT OF TOTAL BITUMINOUS PRODUCTION

State	1929	1930	1931	1932
West Virginia	25.9	26.0	26.5	27.7
Pennsylvania	26.8	26.6	25.6	24.1
Illinois	11.4	11.5	11.6	10.8
Kentucky	11.3	11.0	10.5	11.4
Ohio	4.4	4.8	5.4	4.5
Indiana	3.4	3.5	3.7	4.3
Alabama	3.4	3.3	3.1	2.5
Virginia	2.4	2.3	2.5	2.5
	89.0	89.0	88.9	87.8

TABLE III
RELATIVE PRODUCTION OF BITUMINOUS COAL
IN PRINCIPAL COUNTIES OF INDIANA
1929 - 1932
PER CENT OF TOTAL PRODUCTION OF STATE

County	1929	1930	1931	1932
Pike	13.6	16.4	19.1	20.3
Greene	7.9	10.5	11.4	13.0
Vigo	21.6	18.0	14.4	11.8
Knox	8.0	8.7	10.2	11.7
Sullivan	16.1	16.8	15.8	11.5
Vermillion	12.0	17.2	10.1	7.9
	79.2	87.6	81.0	76.2

TABLE IV

GEOLOGICAL CLASSIFICATION OF PRINCIPAL
COAL SEAMS IN INDIANA

Name of Coal Seam	Formation	Series
Aldrich Friendsville Parker	Post-Allegheny	Pennsylvanian
9th Vein 8th Vein 7th Vein 6th Vein 5th Vein 4th Vein 3rd Vein	Allegheny	Pennsylvanian
2nd Vein Minshall Upper Block Lower Block 1st Vein	Pottsville	Pennsylvanian

TABLE V
AVERAGE PROXIMATE ANALYSES OF INDIANA COALS

AS RECEIVED BASIS

DISTRICT COUNTY SEAM	MOISTURE	ASH	VOLATILE MATTER	FIXED CARBON	SULPHUR	BTU/LB
Princeton-Ayrshire District						
Pike County - 5th Vein	9.8	9.6	36.9	43.7	3.8	11430
Gibson County - 5th Vein	6.9	9.1	38.2	45.8	3.6	12220
Sullivan-Linton District						
Sullivan County						
4th Vein	12.2	7.5	34.6	45.7	1.4	11600
5th Vein	11.0	8.4	36.1	44.5	2.5	11750
6th Vein	12.1	10.1	35.3	42.5	2.2	11215
Greene County - 4th Vein	12.2	7.2	36.3	44.3	1.4	11635
Brazil-Clinton District						
Vigo County						
3rd Vein	11.6	9.0	38.3	41.1	4.0	11525
4th Vein	13.3	8.0	33.6	45.1	1.2	11435
5th Vein	10.7	9.4	39.7	40.2	3.0	11660
6th Vein	13.4	7.3	36.7	42.6	2.2	11450
Minshall	13.1	8.3	36.8	41.8	2.6	11480
Vermillion County						
3rd Vein	10.8	10.3	37.9	41.0	3.6	11400
4th Vein	14.7	7.3	34.2	43.8	1.1	11330
5th Vein	11.6	9.0	39.4	41.0	3.2	11745
Minshall	12.3	11.2	38.0	38.5	4.8	10990
Owen County - Upper Block	12.7	3.4	36.5	47.4	0.6	11950
Clay County - Lower Block	12.3	3.6	35.3	48.8	1.0	12260
Knox District						
Knox County - 5th Vein	10.3	10.2	37.2	42.3	3.5	11530
Boonville District						
Warrick County - 5th Vein	9.4	9.2	38.7	42.7	3.7	11620
Evansville-Newburg District						
Vanderburg County - 5th Vein	9.7	10.2	35.6	44.5	2.9	11440

that may be easily broken along the horizontal partings into plates. The coal itself is not as hard as western Kentucky and Illinois coals but when cleaned and prepared in the smaller sizes, it presents a neat appearance and stands handling well. The block coal, as the name indicates divides into more pronounced blocks than the bituminous, often along surfaces weakened by the presence of mineral charcoal. The cannel coal has the usual dull smooth appearance of cannel coals and breaks with a smooth irregular glass-like fracture.

Uses of Indiana Coal

Much of the production is used within the state for industrial steam generation, as railroad and domestic fuel, and in some cases for gas making. The block coal, because of its low sulphur and ash content, makes a good domestic fuel. The bituminous coal is best burned on traveling grate stokers or in pulverized form. It has a high volatile content and small pieces may be ignited with a match, the coal giving sharp cracking noises as the gas pockets burst and the seams crack open. The sulphur content varies considerably but is largely concentrated in partings that may be cleaned. The fusibility of the ash varies also but is generally below 2200 F. Some low sulphur and high ash fusion coal is found in the Fourth Vein in Greene and Sullivan Counties.

Average proximate analyses on an "as received" basis of coals from the principal seams in the various counties and districts are given in Table V. More complete analyses on a moisture and ash free basis are given in Table VI. The values in Table VI may be calculated to an "as received" basis for any desired moisture and ash content by multiplying by $(1.00 - m - a)$ where m and a are the percentages of moisture and ash, respectively, on an "as received" basis and expressed as decimals. Inspection of the values in Table VI will

TABLE VII

AVERAGE ANALYSIS OF AN INDIANA COAL

	As Received	Moisture Free or Dry	Moisture and Ash Free
Moisture	11.0	-	-
Ash	9.0	10.10	-
Volatile Matter	36.80	41.35	46.0
Fixed Carbon	43.20	48.55	54.0
	100.00	100.00	100.0
Sulphur	2.64	2.97	3.3
Hydrogen	4.48	5.03	5.6
Carbon	64.24	72.19	80.3
Nitrogen	1.28	1.44	1.6
Oxygen	7.36	8.27	9.2
	80.00	89.90	100.0
B.t.u. per lb.	11575	13010	14470

show relatively little difference in the analyses of Indiana coal, and in Table VII is given an average analysis that may be used for an Indiana coal when its analysis is not given and the source of the coal is not known. The analytical values are more or less weighted to approximate the analysis of the bulk of the coal produced in the state.

TABLE VI

AVERAGE ANALYSES OF INDIANA COALS

COUNTY AND SEAM	MOISTURE AND ASH FREE								AS RECEIVED	
	VOLATILE MATTER	FIXED CARBON	SULPHUR	HYDROGEN	CARBON	NITROGEN	OXYGEN	BTU/LB	MOISTURE	ASH
5th Vein										
Gibson	47.4	52.6	4.7	5.5	79.6	1.7	8.5	14475	6-9	9-12
Knox	47.1	52.9	4.2	5.5	79.5	1.5	.9.3	14510	9-11	8-12
Sullivan	44.2	55.8	2.4	5.6	81.0	1.8	9.2	14425	8-15	5-14
Vanderburg	44.5	55.5	3.6	5.4	79.3	1.7	10.0	14280	9-11	10-12
Vermillion	49.6	50.4	4.0	5.8	79.7	1.6	8.9	14520	9-12	8-11
Vigo	49.4	50.6	4.0	5.8	80.2	1.4	8.6	14585	9-12	7-11
Warrick	48.5	51.5	6.0	5.4	78.3	1.6	8.7	14330	6-9	9-11
AVERAGE	47.2	52.8	4.1	5.6	79.7	1.6	9.0	14445	7-12	7-12
4th Vein										
Greene	45.9	54.1	1.7	5.6	81.1	1.8	9.8	14530	11-14	6-10
Sullivan	44.0	56.0	1.9	5.6	81.1	1.8	9.6	14520	9-15	5-10
Vermillion	43.8	56.2	1.4	5.5	81.3	1.8	10.0	14510	13-15	7-9
Vigo	41.3	58.7	0.9	5.7	83.0	1.8	8.6	14560	12-14	6-9
AVERAGE	43.8	56.2	1.5	5.6	81.6	1.8	9.5	14530	9-15	6-10
6th Vein										
Sullivan	44.5	55.5	3.1	5.6	80.1	1.6	9.6	14420	10-15	8-12
3rd Vein										
Vermillion	48.1	51.9	4.6	5.8	79.4	1.5	8.7	14450	9-12	7-14
Vigo	48.5	51.5	6.3	5.6	78.2	1.6	8.3	14490	9-10	10-12
AVERAGE	48.3	51.7	5.4	5.7	78.8	1.6	8.5	14470	9-12	7-14
Minshall Seam (block)										
Vermillion	49.6	50.4	6.3	5.6	77.6	1.5	9.0	14360	10-13	10-12
Vigo	46.9	53.1	3.3	5.8	80.7	1.6	8.6	14620	13-14	7-9
AVERAGE	48.3	51.7	4.8	5.7	79.2	1.6	8.7	14490	10-14	7-12

Combustion Control

The Prime Movers Committee of the Edison Electric Institute addressed a questionnaire recently to a number of operating companies seeking opinions on various phases of automatic combustion control. The replies from eighteen companies are tabulated in the 1935 Report on "Combustion" which has just been issued. These replies are summarized in a statement by the sub-committee as follows.

1. Usual application of automatic control is to raw-coal feeders, forced and induced-draft fans and dampers. Stoker speed control has been applied in six stations reporting, while two stoker-fired stations have omitted stoker speed control and one stoker-fired station has not stated whether or not stoker speed is controlled. No chain-grate operated station has reported.

2. Opinion differs as to whether automatic combustion control saves labor when applied to larger plants, one company reporting definitely that there is considerable saving while another reports no saving. One company reports that if all operations now performed largely automatically had to be resumed manually, the additional physical duties imposed on the present operators probably would result in loss of plant efficiency due to inability to adjust equipment continuously as required for best results.

3. One company reporting on a stoker-fired plant states that the largest factor affecting plant efficiency is automatic control of coal supply. This company states that coal supply manually cannot be regulated as satisfactorily.

4. Most companies report that maintenance is low or reasonable on all parts of the equipment. One company reports that 90 per cent of control maintenance has been on that part applying to stoker speed. One company reporting on an air-operated system points out the necessity of supplying the system with clean dust-free air.

5. The average automatic combustion control equipment is capable of handling the boiler unit from about 25 per cent to 100 per cent of full load. Two companies report automatic control used, satisfactorily, as low as 17 per cent of full load while two others set 20 per cent as the lower limit. On the other hand, two cases are reported with the lower load limit of 40 per cent of maximum. With one exception, all installations are capable of handling the boiler up to 100 per cent of maximum output.

6. Nine companies state that the automatic control cannot be used to bring a banked boiler on the line. One company reports that it can be used if banked fuel bed is in fair condition. Four companies state that with minor manual adjustments and operations, the control probably could be used to bring on banked boilers. It seems, however, universal practice among companies reporting to bring banked boilers up manually to minimum rating mentioned in 5, before automatic control is cut in.

7. All companies agree that greatest value of automatic control is realized on swing boilers, although it is very important on base load boilers as well. In the majority of cases reported, the automatic control varies load on all boilers although it is selective and can be used to swing one boiler or a group of boilers in certain cases. The choice of whether load swings are handled on one or more boilers seems to be a matter of plant or system conditions rather than limitations of control equipment.

8. The only general conclusion that can be drawn from the statements regarding objectionable features is that these are complication and high cost. Difficulty in maintaining oil piping tight on certain types of systems seems to be common. One company reports that inaccessibility of parts and assembly adds to cost of maintenance. A case has been reported of some difficulty experienced with coal feed regulation on a bin-and-feeder system. Coal feed is controlled by feeder speed. Varying quantities of moisture affect the amount of pulverized coal that will be fed at any given speed, making frequent manual adjustments necessary.

Mercury switches have caused one company trouble on account of contamination of mercury and apparent over-rating in current capacity.

One company reports that range of steam pressure on automatic control is possibly too narrow, as applied to a 1400 lb per sq in. plant. If pressure should vary 35 to 40 lb per sq in. either way, conditions must be restored by hand as it is beyond the range of automatic control.

Production of Electricity

The total production of electricity for public use in the United States in August was the largest ever produced in that month according to statistics compiled by the U. S. Geological Survey. The average for the month was 277,000,000 kwh per day, an increase of 2.6 per cent above the average daily production in July, which is about the normal change from July to August. The September figures have not yet been compiled but are expected to exceed those of August.

The average daily production of electricity by the use of water power in August was less than in July, indicating the beginning of the seasonal decrease in output by water power due to decreasing stream flow. The heavy precipitation early in September may tend to reduce the usual seasonal decrease in the flow of streams to some extent.

TOTAL MONTHLY PRODUCTION OF ELECTRICITY FOR PUBLIC USE

	1934 Million kilowatt-hours	1935 Million kilowatt-hours	Per cent change 1934–35	Produced by water power (per cent) 1935
January	7,651	8,349	+ 9	39
February	7,066	7,494	+ 6	40
March	7,735	8,011	+ 4	44
April	7,458	7,817	+ 5	46
May	7,704	8,021	+ 4	46
June	7,490	7,873	+ 5	44
July	7,617	8,372	+10	43
August	7,722	8,586	+11	39
September	7,207			
October	7,833			
November	7,609			
December	8,058			
	91,150			

L. V. Sanderson has been appointed sales representative for the American District Steam Company in Philadelphia and the State of Delaware with headquarters at 1108 Otis Building, Philadelphia, Pa.

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

British Station Installing 1900-Lb Boilers

What will be the highest pressure installation in England is described in the September 19 issue of *The Electrical Times*. This refers to the rebuilding of Brimsdown "A" Power Station of the North Metropolitan Electricity Supply Company, London, for which two 210,000 lb per hr Loeffler boilers have been ordered. These will generate steam at approximately 1900 lb pressure, 940 F total temperature and will supply a 50,000-kw turbine-generator. The high-pressure element of this turbine, of 20,000 kw, will exhaust at 195 lb to both live-steam and flue-gas reheaters, from which the steam will be returned to the 30,000-kw low-pressure element. Inasmuch as a Loeffler type boiler must be started by an auxiliary supply of steam some of the newer existing 150-lb boilers will be left for this purpose, also for supplying some steam to the low-pressure element of the turbine in an emergency.

These two boilers will be constructed in England by the Mitchell Engineering Ltd., which has the British rights for the Loeffler boiler.

Superposition in the Marine Field

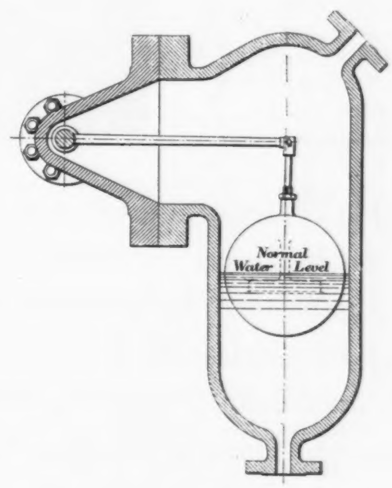
Under the title, "Modern Marine Steam Progress," J. S. Gander, in the September issue of *Engineering and Boiler House Review* describes the installation of a Sulzer mono-tube steam boiler in the cargo liner "Kertosono" of the Rotterdamsche Lloyd Line. This unit will operate at 880 lb pressure, 705 F total steam temperature and will supply steam to a high-pressure turbine that will exhaust to the present low-pressure plant consisting of Scotch boilers and a Parsons turbine operating at 200 lb. One of the original five Scotch boilers has been removed to make room for the new steam generator and another will be employed to provide steam for auxiliaries. The superimposing of the high-pressure unit will increase the shaft horsepower from 4500 to 5800 and is expected to increase the speed of the vessel from 13 to 15 knots.

Errors in Coal Analysis

Zeitschrift des Vereines deutscher Ingeneuire reviews the report of a committee appointed by the V. D. I. to consider revising the rules for acceptance tests of steam boilers. It was the opinion of this committee that the principal probable error, which must be taken into account when specifying tolerances, lies in determination of the calorific value of the fuel. This view was substantiated by comparison of analyses from sixteen laboratories which varied to an extent that would account for 3 per cent in boiler efficiency. As a result V. D. I. has prepared special rules for sampling and arranged for maintaining a list of approved laboratories.

New High- and Low-Level Alarm

Engineering of September 20 reports the recent Engineering and Machinery Exhibition at Olympia. Among the numerous exhibits described is a new high- and low-water alarm for steam pressures up to 900 lb which has been bought out by Hopkinsons Ltd. This alarm is external to the boiler and contains a spherical float of alloy steel, the lever of which is attached to a



Section through alarm

cylindrical spindle mounted eccentrically in knife-edge bearings. Rotary movement of the spindle causes the valve to uncover or cover small holes in the seat leading to two whistles. When the maximum high-water level is reached steam is admitted to the high shrill-note whistle and when low water is reached steam is admitted to the other whistle which gives a deep resonant note. The seats are held in place between flanges and can be set to adjust the water level as required by rotating them in accordance with a graduated scale.

H_2CO_3 Corrosion in High-Pressure Boilers

"With the exception of distilled water, every water treated by the customary softening methods in the past has contained carbonate, as a result of the addition of excess soda, whether it was freed of its hardness by chemical softeners or by an exchange of bases," says *Archiv für Warmewirtschaft und Dampfkesselwesen of* May 1935.

In late years it has been recognized from tests in Germany that in high-pressure boilers H_2CO_3 corrosion may occur under certain conditions but that such corrosion is not to be feared in boilers operating at less than 750 lb working pressure, or that operation at 1500 lb may be free from such corrosion if the feedwater is free of carbonates. It appears that the CO_2, freed by splitting up soda, combines with water to form H_2CO_3 and at temperatures of about 500 F the iron is attacked to form iron carbonate and free hydrogen, both of which spread rapidly over the boiler surface in the form of very small rhomboids that are interlocked and appear to penetrate the metal. The color is essentially yellowish-white. The outwardly appearing grey-brown coloring comes from the very small amounts of ferrous and ferric oxides present.

From these observations it was concluded that boilers operating at more than 750 lb pressure should be fed only with water having a content of fixed carbonate not in excess of 10 mg per liter of CO_2. A complete soda separation does not occur.

Conceding that many boilers have operated almost continuously at pressures of 700–900 lb without showing evidence of such formations, the author attributes this to the protective action of caustic soda and sodium sulphate or phosphate.

Properties of Low-Nickel Steels Containing Manganese

The mechanical properties of steels containing 0.3 to 0.4 per cent carbon, 0.7 to 1.35 per cent manganese and 0.5 to 2.0 per cent nickel are discussed in a paper by R. H. Greaves before the September 1935 meeting of the *Iron and Steel Institute* of Great Britain.

The amount of nickel which may be added to a steel containing manganese without inducing a tendency to air-hardening decreases from well over 2 per cent with a manganese content of 0.7 per cent down to 1 per cent when the manganese content is 1.35 per cent. In the normalized condition a steel containing 0.35 per cent carbon, 1 per cent manganese and 1 to 1.5 per cent nickel gives good mechanical properties, better than those previously obtained from 3 per cent nickel steel, 1.4 per cent manganese steel and silico-manganese high elastic limit steels. Steels of approximately this composition do not harden fully when quenched in oil in sections ¾ in. thick, and are subject to mass effect, but their mechanical properties in the oil-hardened and tempered condition are equal to those for which 3 and 4 per cent nickel steels of similar carbon and low manganese contents are frequently employed.

An increase in manganese and nickel, either separately or together, within the limits investigated, diminishes the mass effect and leads to an improvement in the mechanical properties without serious loss of ductility. It also increases the range of temperature from which quenching may be carried out without adversely affecting the mechanical properties. Steels of the higher manganese and nickel contents are susceptible to temper-brittleness, but air-cooling of sections ¾ in. thick does not give rise to a low impact figure. All the steels investigated hardened fully on water-quenching, and the relatively high manganese content of some of them did not result in any cracking.

It would appear that, in suitable circumstances where a yield point of over 32 tons per sq in. is required, with maximum load over 45 tons per sq in. and impact figure over 40 ft-lb, oil-hardened and tempered steels containing 0.35 to 0.40 per cent of carbon and 1.0 to 1.3 per cent of manganese with about 1 per cent of nickel may be used with economic advantage in sections up to $2^1/_2$ in. in thickness in place of 3 per cent nickel steel. If a high impact figure is required and the section to be treated exceeds $2^1/_2$ in. in thickness, it may be necessarv to reduce the carbon to 0.30 and increase the nickel to 1.5 or even 2.0 per cent, since steels with the higher carbon content indicated above, when treated in large sections, give an impact figure below 40 ft-lb.

Concrete Cooling Towers

Because of the location of many power stations at sites removed from large bodies of water suitable for condensing purposes, cooling towers have long been extensively used in England. For this purpose concrete towers are coming into favor and have superceded those of wood especially for towers over a certain capacity. B. Frisby in the September issue of *The Power and Works Engineer* discusses this trend and points out that an outstanding advantage of the concrete tower is its durability compared with the wooden tower and that the small maintenance over a period of years more than offsets its greater initial cost. Furthermore, air infiltration is avoided in the concrete tower and a better draft is available owing to the greater height to which they can be built, and the hyperbolic design makes dead space negligible.

The largest of such towers constructed in England to date are those serving the Hams-Hall Station at Birmington with a capacity of five million gallons per hour. Other outstanding examples cited by the author are two towers for the Croydon Station, the latest of which has an hourly capacity of 1,500,000 gallons, is 140 ft in diameter and 184 ft high.

Oil from Coal

The Fuel Economist (London) for September describes the new large plant of the National Coke & Oil Company at Tipton which is about to start commercial operation for the production of oil from coal. It includes a briquetting plant for handling the fines screened from the residual coke.

The process is one of destructive distillation which yields as final products only coke and light oils, the heavy oils and tars which occur in the initial distillation of the coal are re-cycled and crushed. The raw coal is treated in pulverized form admixed with about 50 per cent heavy oil from the carbonization of previous charges. A rotating horizontal cylindrical steel retort, without refractory lining, is employed and is externally fired.

With abundance of coal but no local oil resources England is obviously receptive to processes for obtaining oil from coal.

Recording of Turbine Blade Erosion

By ROBERT P. HOOPER, Testing Engr.,

Brooklyn Edison Company, Inc.

EROSION of turbine blades caused by the formation of moisture in the steam in the low-pressure ends of turbines increases with increased efficiency of turbine design. In recent years the turbine manufacturers have attempted to reduce the increasing rate of erosion by the use of erosion resistant alloys for turbine blades and by the improvement of stage drains for removing the moisture as it is formed. Despite these efforts erosion of the blades on the low-pressure end of steam turbines is not uncommon.

Fig. 1—Photographing turbine blades to obtain profile record

Several methods are in use for following the progress of blade erosion. One method consists of comparing successive plaster casts of certain typical blades. While satisfactory results can be obtained by this method it is rather cumbersome to use. A preferable method consists in obtaining an accurate profile of individual blades by the use of blueprint paper or sensitized photographic paper. The photographic paper has the advantage over the blueprint paper in that it gives a sharper outline of the blade and requires considerably less time of exposure. The paper, which is preferably a contrast paper of the type "Azo F 4," is held in close contact with the blade by a sheet metal form moulded to the shape of the blade. With the turbine spindle removed exposure is made by any convenient type of electric lamp, but with the spindle in place it was found that a tubular lamp of a size which could be inserted between the blades gave most satisfactory results. When using a 25-watt lamp an exposure of from 10 to 15 sec has been found sufficient.

Fig. 2—Typical profile record of eroded turbine blade

Fig. 1 illustrates the procedure of taking the photographs and Fig. 2 shows a photographic record (positive) of erosion of the leading edges of blades in the last wheel of a large modern turbine. The blade material is stainless steel. By taking these photographs periodically of identical blades it is possible to compare the photographs from time to time and thus follow the progress of erosion.

H. M. Cushing of the Buffalo General Electric Company was probably the first to suggest and use blueprint paper for this purpose.

Oscar Junggren Dead

Oscar Junggren, who was associated with the development of many of the most important advances in the design of large steam turbines and who until recently was design engineer of the General Electric Company's turbine department, died on September 24 at his home in Schenectady, N. Y., following an illness of about fourteen months.

A prolific inventor, he had some 130 patents to his credit, most of which were related to new developments in the steam-turbine field. Indicating his fertility of imagination is the fact that, during the eight years from 1924 to 1932, thirty turbines, each representing an entirely new design and marking an outstanding achievement in the field, were developed under his direction—an average of one new turbine design nearly every four months.

Mr. Junggren was born in Landskrono, Sweden, January 10, 1865. He attended the engineering college of Malmo in that country, graduating in 1885 from the mechanical engineering course. Coming to this country, he was first employed by the Edison Electric Company in New York City in 1889. In 1891 he became associated with the General Electric Company, working on the design of steam power equipment in Schenectady, and continued in this general field throughout his long career.

NEW CATALOGS AND BULLETINS

Any of these publications will be sent on request.

Air-Operated Combustion Control

A 32-page bulletin entitled "Air-Operated Combustion Control" and designated as No. 102, is being distributed by Bailey Meter Company. This describes a complete combustion control system which automatically maintains steam pressure, combustion efficiency and furnace draft. It explains in detail how the system ties in with the company's boiler meter to automatically readjust fuel-air ratio. Diagrams and photographs illustrate how the system may be applied to boilers ranging in size from 200 hp up, and fired either with fuels in suspension or with stokers.

Blow-Off Valves

Yarway forged-steel blow-off valves for 600 to 1500 lb working pressure of the single and tandem type are described in a new bulletin issued by Yarnall-Waring Company. A table of dimensions for various sizes of valves is included.

Blowers

Bulletin 21-B-17 descriptive of Victor-Acme rotary positive blowers has just been issued by Roots-Connersville Blower Corporation. The power economies that are possible when moving varying volumes of air by the blowers, are illustrated by a diagram. Other operating characteristics of known, controlled volumes, positive air movement, freedom from internal seal or lubricant, are described, along with illustrations of the Types "SB" and "AFS" designs. A selection table, giving net capacities at listed speeds and pressures, is included.

Boiler Plant Equipment

"Boiler Plant Equipment" is the title of a four-page bulletin just issued by Combustion Engineering Company Inc. which describes, in condensed form, all the types of boilers, complete steam generating units, stokers, pulverized fuel systems and heat recovery equipment furnished by that company. Line drawings of such equipment are included.

Calorimetric Testing Kits

A circular has just been issued by National Aluminate Corporation describing the "Nalco" calorimetric Testing Kit. While these kits have been used in testing by that company for some time, they are only now being offered for sale. The circular describes the construction of the kits, discusses calorimetric methods of testing and gives prices of equipment and solutions.

Cinder Reclaiming and Fly Ash Disposal

United Conveyor Corporation has a new catalog describing its "Steamatic System" for (1) returning cinders by pneumatic transport either directly to the furnace or to the bunkers, the flow of air being produced by a motor-driven air pump or a multi-jet steam exhauster; and (2) the handling of fly ash to storage tanks or to the ashpit when the quantity is small. Diagrams showing various schemes for disposal are included as well as photographs of numerous installations.

Electrodes

Murex heavy-coated electrodes for welding are covered in a well-illustrated 16-page catalog issued by Metal & Thermit Corporation. The welding of mild commercial steel and various new alloy steels is discussed and a section is devoted to repair and maintenance welding.

Feedwater Heaters

Feedwater heaters of the deaerating type, built to meet individual plant requirements, are described and illustrated in a bulletin just issued by Worthington Pump and Machinery Corporation. Heaters of both cylindrical and rectangular cast-iron plate design are included.

Fuel-Burning Practice

"Industrial Coal-Burning Equipment" is the title of a 20-page bulletin issued by Appalachian Coals, Inc. This analyzes a recent survey by the Market Research Department of that company covering 7284 industrial steam plants in eight states. It summarizes by states and as totals the number and types of fuel-burning equipment, the average horsepower per unit of each type, the total annual coal burned and the coal per horsepower. Similar information is included as to boilers. The states covered are Georgia, Indiana, central and eastern Kentucky, lower Michigan, North Carolina, western Ohio, South Carolina and western Tennessee. The compilation presents what is believed to be a representative picture of industrial boiler and fuel-burning practice. A detailed study of the tabulations presents a wealth of information that should be most useful in fuel market studies.

Steam Accumulator

Bulletin RA-35-5, descriptive of the Ruths steam storage systems, has recently been released by Foster Wheeler Corporation. This new bulletin includes charts and illustrations showing that when the steam production exceeds plant demand the surplus is stored in the steam accumulator, by condensing it in the water contained in the accumulator. When the processing departments of the plant require more steam than the boilers are producing, at their fixed rate, the automatic valves on the accumulator open and release steam for process work at the pressure required in the process departments. This arrangement is shown diagrammatically in the bulletin and examples are given of installations, and actual flow charts show the ability of the accumulator to absorb and release steam as required.

Steam-Driven Compressor

The Ingersoll-Rand steam-driven air or gas compressor (Type XPV) which is a new machine in everything but name, is described in detail in a new 48-page profusely illustrated catalog. Boiler plants today are seldom designed primarily to furnish steam for reciprocating engine-driven machinery. On the contrary, the steam-driven compressor is usually chosen to fit into the existing steam-plant conditions with a view to maintaining the proper steam balance of the plant as a whole. For this reason the "Type XPV" compressor has been developed around a standardized set of frames and running gears. For each of these there has been developed a wide range of sizes of steam cylinders to take care of varying steam requirements. Likewise, in addition to the standard two-stage 100-lb compressing end for each of these frames, a wide range of sizes and types of compressor cylinders is available to take care of special compressing problems. Many combinations of governing and regulating equipment have been developed to insure proper regulation under every possible set of conditions. These are discussed.

Stoker Unit

The new "C-E Stoker Unit," applicable to all types and sizes of boilers from small heating units to power boilers developing up to 400 hp, is described in a folder recently issued by Combustion Engineering Company, Inc. This unit is made in both the center- and side-retort arrangements and employs a combination of screw and reciprocating raw feed for the coal, alternate fixed and moving grate bars having ports through which air from the integrally mounted fan is admitted. A constant speed motor is employed to drive the stoker but by means of a variable speed transmission sixteen rates of coal feed are attainable.

The new stoker unit reflects Combustion Engineering Company's long and vast experience in designing, manufacturing and installing stokers for every type of coal and practically all types of boilers, and particularly its 25 years' experience in the production of single-retort underfeed stokers.

The folder, besides clearly illustrating all details of the stoker unit, also shows a number of representative installations that have been made in factories, laundries, greenhouses, dairies, hospitals, churches, schools, apartment houses, hotels, warehouses, etc.

Zeolites

The mining, processing, manufacture and uses of zeolites are covered in a 34-page booklet issued by The Permutit Company. The text is based on a paper by Eskel Nordell of the technical department of that company which was presented at the Michigan Conference on Water Purification, but the 84 illustrations include many that were subsequently added.

To Promote Lubrication Engineering as a Career

In response to a demand for some organized method of supplying authentic information on industrial lubrication problems, there has been established the Industrial Lubrication Council, Inc., a non-profit organization for educational purposes with headquarters at 51 East 42nd Street, New York. The aims and objectives of the Council, may be summarized as follows:

1. Advancement of the art and science of machine lubrication.
2. To provide adequate equipment and personnel to conduct general education on the fundamentals and practice of machine lubrication.
3. To develop and establish in the technical and trade schools and colleges the profession of lubrication engineering.
4. To stimulate throughout general industry a greater interest in fine lubrication and the acceptance of the lubrication engineer as a member of the engineering staff of industrial plants.
5. To provide reasonable and sound technical material in the form of special articles, textbooks and bulletins for the continual training of the lubrication engineer in general industry and his information as to the latest thought and practice and experiments under way.
6. To establish and maintain a clearing house for the exchange of ideas on all problems of application and research on the broad subject of lubrication for the benefit of the membership of the Council and educational institutions.
7. To conduct conferences and meetings and to provide an open forum for discussion of specific lubrication problems, to provide and give lectures and to provide facilities for demonstrating whatever is new in the field of machine lubrication.

Due to the economic changes of the past decade, there has been a general extended employment by the manufacturing industries of the plant lubrication engineer, a trend well along. These men, as a rule, are used for the purpose of reducing the cost of lubrication by making the cheapest oils obtainable do the work in order to justify their employment. There has been no method of instructing these industrial lubrication engineers on the finer points of lubrication. They have no uniform manner of handling their problems.

When general industry demands lubrication engineers to be members of their maintenance staffs, they should be able to secure through the regular channels men of sufficient initial training to handle the usual lubrication problems presented by every plant. There has never been a definite plan for the uniform training of lubrication engineers, due to the lack of authorized textbooks on the subject.

Uniformity of lubrication practice does not and never has existed. Therefore, when it comes to the method of extending the use of lubrication engineers throughout general industry where several thousand men can be employed in the future, it is necessary to establish some uniform manner of teaching the fundamentals.

There has been prepared a standard textbook or course of instruction in lubrication engineering for the technical and trade schools, colleges and universities so that engineering students can be properly trained for future employment in this branch of mechanical engineering.

These advantages will be extended also to the engineers who are now employed in the plants and shops of industry. This will be done through a correspondence instruction course along with a regularly issued series of information bulletins on the latest practical developments in the many subjects covered by "lubrication," and finally by a handbook, the forty-five chapters of which thoroughly cover the modern thought and practice on this important subject.

EQUIPMENT SALES
Boiler, Stoker, Pulverized Fuel

As reported by equipment manufacturers of the Department of Commerce, Bureau of the Census

Boiler Sales

Orders for 85 water-tube and h.r.t. boilers were placed in August

	Number	Square Feet
August 1935	85	253,313
August 1934	93	365,642
January to August (inclusive, 1935)	649	2,095,990
Same period, 1934	620	1,835,488

NEW ORDERS, BY KIND, PLACED IN AUGUST 1934–1935

	August 1934		August 1935	
Kind	Number	Square Feet	Number	Square Feet
Stationary:				
Water tube	58	305,266	56	218,463
Horizontal return tubular	35	60,376	29	34,850
	93	365,642	85	253,313

Mechanical Stoker Sales

Orders for 269 stokers, Class 4* totaling 47,355 hp were reported in August by 68 manufacturers

	Installed under			
	Fire-tube Boilers		Water-tube Boilers	
	No.	Horsepower	No.	Horsepower
August 1935	193	24,298	76	23,057
August 1934	191	25,002	50	18,891
January to August (inclusive, 1935)	875	116,866	374	143,044
Same period, 1934	841	110,991	338	132,578

* Capacity over 300 lb of coal per hr.

Pulverized Fuel Equipment Sales

Orders for 17 pulverizers with a total capacity of 79,550 lb per hr were placed in August

STORAGE SYSTEM

	Pulverizers				Water-tube Boilers		
	Total number	No. for new boilers, furnaces and kilns	No. for existing boilers	Total capacity lb coal per hour for contract	Number	Total sq ft steam-generating surface	Total lb steam per hour equivalent
August 1935	..	..	..	....	..	....	......
August 1934	..	..	..	....	..	....	......
January to August (inclusive, 1935)	..	..	..	....	..	....	......
Same period, 1934	2	1	1	46,000	*	*	*

DIRECT FIRED OR UNIT SYSTEM

	Pulverizers				Water-tube Boilers		
August 1935	17	8	9	79,550	15	78,384	762,580
August 1934	21	16	5	143,270	15	79,088	1,107,800
January to August (inclusive, 1935)	79	46	33	369,310	67	371,737	3,369,160
Same period, 1934	69	50	19	456,180	52	350,663	3,762,320
					Fire-tube Boilers		
August 1935	..	..	..	.....	..	.....	....
August 1934	3	2	1	2,400	3	616	23,000
January to August (inclusive, 1935)	3	..	3	3,300	3	6,130	32,500
Same period, 1934	7	2	5	7,200	8	8,116	64,000

* Data not available.

Business Notes

Combustion Engineering Company, Inc., announces the appointment of Wilson Machinery & Supply Company, Inc., Lexington, Kentucky and William Franklin, Buffalo, New York, as representatives of its Industrial Stoker Division for the sale of C-E Stoker Units—a line of underfeed stokers applicable to boilers from the small size heating units to power boilers developing up to 400 hp.

Ernst Water Column & Gage Co., Newark, N. J. announces the appointments of MacDonald Sales & Engineering Co., 6312 Sherwood Rd., Philadelphia as representative for Philadelphia and Baltimore district; also H. C. Hartmann, 3617 Winnebago St., St. Louis, as district representative.

John D. Swain has been elected vice president of Electro Metallurgical Sales Corporation. He has been vice president of The Linde Air Products Company and the Union Carbide Sales Company, which are other units of the Union Carbide and Carbon Corporation and has been actively connected with these units for a period of twenty years.

Bernitz patent No. 1,393,606 and Snow patent No. 1,627,349 have just been held valid and infringed by the United States Circuit Court of Appeal for the Third Circuit in the case of Bernitz Furnace Appliance Company vs. Dillon B. Wilson.

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